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Treasurer.....H. E. Park
Board of Governors: L. S. Maehling, A. F. Metzger, C. H. Schneider

Rocky Mountain

Organized 1944
Headquarters, Denver, Colo.

President.....F. L. Adams
Vice-President.....J. F. Mohan
Secretary.....J. J. Johnson
Treasurer.....H. J. Woehlke
Board of Governors: A. W. Cooper, H. H. Herman, G. D. Maves

St. Louis

Organized 1918
Headquarters, St. Louis

President.....B. C. Simons
1st Vice-President.....J. H. Carter
2nd Vice-President.....B. L. Evans
Secretary.....W. A. Russell
Treasurer.....J. S. Rosebrough
Board of Governors: C. H. Burnap, J. F. Naylor, Jr., W. J. Oonk, H. C. Sharp

OFFICERS OF LOCAL CHAPTERS—1946 (*continued*)

South Texas

Organized 1938
Headquarters, Houston

President.....B. P. Fisher
Vice-President.....D. M. Mills
Secretary.....C. C. Quin, Jr.
Treasurer.....L. L. Ladewig
Board of Governors: R. J. Salinger, J. A. Walsh, W. J. Way, II

Southern California

Organized 1930
Headquarters, Los Angeles

President.....Art. Theobald
Vice-President.....R. A. Lowe
Secretary.....R. S. Farr
Treasurer.....J. L. Blake
Board of Governors: L. B. Davenport, J. S. Earhart, Maron Kennedy, J. M. Moriaty

Southwest Texas

Organized 1946
Headquarters, San Antonio

President.....F. C. Benham, Jr.
Vice-President.....G. R. Rhine
Secretary-Treasurer.....L. S. Pawkett
Board of Governors: R. W. Barnes, F. C. Benham, D. E. Locher, A. J. Rummel, E. P. Weatherby, Jr., I. W. Wilke

Utah

Organized 1944
Headquarters, Salt Lake City

President.....J. T. Young, Jr.
Vice-President.....E. V. Gritton
Secretary-Treasurer.....E. J. Watts
Board of Governors: R. C. Brown, D. B. Holford, H. G. Richardson, Alfred Richeda

Virginia

Organized 1946
Headquarters, Norfolk

President.....R. C. Thomas
Vice-President.....W. H. Webster, Jr.
Secretary.....J. F. Reynolds
Treasurer.....W. G. Hayes
Board of Governors: W. G. Gorsuch, W. P. Robinson, J. E. White

Washington, D. C.

Organized 1935
Headquarters, Washington, D. C.

President.....L. Bert Nye, Jr.
Vice-President.....H. H. Hill
Secretary.....J. N. Stewart
Treasurer.....P. R. Achenbach
Board of Governors: A. E. Beitzell, W. H. Littleford, F. J. Pratt

Western Michigan

Organized 1931
Headquarters, Grand Rapids

President.....J. W. Miller
Vice-President.....Frank Harbin, Jr.
Secretary.....K. E. Robinson
Treasurer.....H. W. Wolters
Board of Governors: H. J. Metzger, C. A. Simonds, J. H. Van Alsburg

Western New York

Organized 1919
Headquarters, Buffalo

President.....Herman Seelbach, Jr.
1st Vice-President.....G. E. Adema
2nd Vice-President.....Edwin Woolcock
Secretary.....F. J. Weber
Treasurer.....B. C. Candee
Board of Governors: M. C. Beman, Joseph Davis, Roswell Farnham, W. R. Heath, F. A. Moesel

Wisconsin

Organized 1922
Headquarters, Milwaukee

President.....E. W. Gifford
Vice-President.....M. W. Bishop
Secretary.....B. M. Kluge
Treasurer.....J. R. Vernon
Board of Governors: A. S. Krenz, O. A. Trostel, J. H. Volk

Student Branch

Organized 1946
Headquarters, Texas A. & M. College
College Station, Tex.

President.....K. W. Ryan, H. M. Kay*
Vice-President.....H. M. Kay
Secretary-Treasurer.....G. H. Jackson
Board of Governors: W. W. Caudill, J. S. Hopper, W. E. Long

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* Filled Unexpired Term.

1282

TRANSACTIONS
of
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1282

FIFTY-SECOND ANNUAL MEETING, 1946
New York, N. Y.

THE 52nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, which was held at the Hotel Commodore, New York, N. Y., January 27-30, was well attended with a registration of 725, of which 514 were members, 134 guests and 77 ladies. During the four day meeting, the members approved a new charter, voted to change the nominating committee procedure, adopted a revision in the method of handling funds, and discussed 10 technical papers.

FIRST SESSION—MONDAY, JANUARY 28, 2:00 P.M.

Dr. C.-E. A. Winslow, president of the Society called the first session to order in the West Ballroom at 2:00 p.m., and introduced H. J. Ryan, New York, N. Y., president of the New York Chapter who welcomed the members and guests to the first post-war meeting of the Society.

President Winslow, in giving his report as president, which follows, referred to the reports of the Council, Council Committees, Secretary, and the Accountant's report and reviewed the activities of the Society for the past year.

PRESIDENT'S REPORT

I shall attempt to remind you briefly of some of the outstanding events that occurred during the year.

The first important item is the membership, the foundation of the Society. At the beginning of the year we started with 3,854 members and the chairman of the Membership Committee announced that 4,500 would be an appropriate goal for the year 1945. I am happy to report that at the last official count the membership totaled 4,504 as shown in the following tabulation:

STATUS OF MEMBERSHIP AS OF JANUARY 1, 1946

Charter Members	1
Honorary Members	2
Life Members	112
Members	2,380
Associates	1,767
Juniors	216
Students	25
Total	4,504

Elections and Reinstatements	735
Resignations, Cancellations and Deaths	85
Net gain for year 1945	650
Membership as of January 1, 1945	3,854
Net gain	650
Membership as of January 1, 1946	4,504

To the chairman of the incoming Membership Committee, I would like to say that 4,600 will by no means be an adequate goal for the present calendar year, we should aim for 5,000!

In order to simplify and expedite the admission of candidates for membership the Committee on Admission and Advancement has recently tried out a new procedure of meeting once in every four to six weeks. The committee members go over all the applications together, providing a better basis for judging the merits of each applicant and thereby saving a good deal of time in processing the applications.

Though the membership has increased to a considerable extent there have been no new chapters formed during the year; we still have 37 chapters in the United States and Canada. I believe that you will be interested in the statistics regarding chapter membership which you will find in the printed notice of the amendments to the Constitution. The membership of the chapters varies widely from eleven in Oklahoma to 225 in New York.

I was personally very much impressed by the vitality of the chapters in the cities which I visited during the year. I think the difference between 1944 and 1945 was notable in the growth and realization of a sense of responsibility on the part of the chapters. G. D. Winans, chairman of the Chapter Relations Committee, has contributed to this materially through the News Letter that he has been sending out. In addition, many guest speakers have been sent to the local chapters by the Chapter Relations Committee, and the officers have visited all but three of the chapters.

Other Society activities have not progressed as favorably. We have had great difficulties during the war period and thereafter. You know the phrase when anything goes wrong—*Haven't you heard there's a peace on?* Many difficulties have arisen in connection with arrangements for meetings, and I regret to say that we still are not in a position to announce the place of the Semi-Annual Meeting, 1946. Arrangements, however, have definitely been made for a meeting for January, 1947, in Cleveland, Ohio, at which time the Exposition will be resumed, and it is the hope of the Council that thereafter it will be an annual instead of a biennial event.

The principal problem before the Council this year has been that of finding permanent quarters for the Research Laboratory. Temporary quarters in Cleveland were occupied by the Laboratory which had to be vacated by the end of March, 1945. The special committee under E. N. McDonnell's chairmanship, with the Executive Committee and the Finance Committee, devoted an enormous amount of time to this problem, visiting 22 places in Cleveland, other cities having also been carefully considered. Problems of leasing as well as of buying were also finally investigated. Fortunately we were able to locate a piece of property suitable for our purpose on a very well-situated lot on Euclid Avenue. An old-fashioned frame house is located in front of the property behind which is a very large two-story brick auditorium that is ideally suited for use as a laboratory. Due to the various local circumstances we were able to purchase this property on very favorable financial terms. We have had a lot of talk about permanent homes. Well, our permanent home will be in heaven, we hope, or somewhere else. We do not claim that this is a permanent home but it is a place in which the work of the Laboratory can go forward actively and successfully, with an assurance that when we move, we will move because we want to and not because we are turned out. Mr. McDonnell particularly deserves the greatest credit for his tireless devotion to this task.

Under the direction of Prof. G. L. Tuve, the chairman of the Committee on Research, and under the leadership of Mr. Tasker, the work of the Laboratory is in full swing. There are nine laboratory projects for cooperative research under contract with ten different Universities. We still have some 20 technical advisory committees aiding the research work of the Laboratory. Another fortunate circumstance in connection with the Laboratory has been the assignment by the United States

Public Health Service of one of its very able experts, Dr. Allen D. Brandt, a specialist on industrial hygiene, to work in the Laboratory in Cleveland. The Public Health Service guarantees his salary and that of his assistants, and the Laboratory is furnishing the quarters and the apparatus. It is an opportunity, I think of great promise for the Society to effect a correlation with medical and engineering knowledge in this important field.

The matter of financing the Laboratory has received the major share of our attention. The research funds that the Society receives from dues, and from the proceeds of the Exposition and the publications, in addition to the gifts that are received in response to letter appeals to the industry, amount to somewhere in the neighborhood of $40,000 or $50,000 a year. A minimum budget, we feel, for research, should be well over $100,000.

The Council has devised a plan by which this money can be obtained. A special committee has been appointed under the chairmanship of Dr. A. C. Willard, president of the University of Illinois, and it has developed a program which contemplates an appeal to a smaller number of industries not on the ground of charity, not on the ground of doing something for somebody, but on the ground that this Laboratory of the Society should be the Research Laboratory for the profession and industry. Obviously the character of the work will not change; the Laboratory will not undertake any commercial work, nor any testing of materials. We are convinced, however that many industries have basic problems, problems of interest to all, that they want solved.

The plan is to make no general appeal for funds of this type, but rather to make a careful approach to selected industries for gifts to support earmarked research along lines of fundamental importance on which they desire light. We are very confident that this procedure will be successful. We have been waiting something like six months to get the right man to act as the agent to carry out the fund-raising campaign under the leadership of this committee, and I am very happy to say that we have found an ideal man for that task, C. A. McKeeman. Mr. McKeeman is familiar with research, has been connected with the research program of the Society, and has admirable gifts in interpreting his knowledge to business executives.

In closing I would like to review some of the changes which have been made on the HEATING, VENTILATING, AIR CONDITIONING GUIDE, which has been quite radically revised. Some 100 pages have been added and as a result of these additional pages we shall not be able to get out as large an edition as we otherwise might have wished to do. C. S. Leopold, the chairman, has done a splendid job on this. I ought to warn you that the committee this year has changed the order of a number of chapters in THE GUIDE and my prediction is that we shall hear a good many complaints about that.

These, I think, were the major events of the year.

Respectfully submitted,

C. E. A. WINSLOW, *President*

REPORT OF COUNCIL

The business affairs of the Society were handled by the Council, which held four meetings, January 24 in Boston, May 14 in Chicago, October 7, 1945, in Cleveland, and January 27, 1946, in New York City.

At the organization meeting in Boston the Council approved the appointment of committees suggested by President Winslow, and confirmed the appointment of the Secretary and Technical Secretary, selected depositories for Society funds, and appointed a certified public accountant.

The Council agreed to sponsor the organization of the Joint Committee on Psychrometric Data and voted to increase the personnel of the Guide Publication Committee because of the heavier volume of work required. It was voted that the customary summer meeting would not be held.

Immediately following the Annual Meeting the Society suffered an unexpected loss in the death of Comdr. F. C. Houghten, former Director of the Society's Research Laboratory and again in May the Society lost its Second Vice-President, W. A. Russell, Kansas City. Since that time it has also suffered the loss of three past presidents, Thornton Lewis, D. S. Boyden and C. B. J. Snyder, and very recently a

former Treasurer of the Society, F. D. Mensing. Appropriate resolutions have been drafted on behalf of the Council and Officers.

The five Council Committees have been exceedingly active this year and several special Council Committees were appointed, one to solve the problem of Research Fund Raising, another the Housing of Society Offices and Laboratory, and third to assemble the History of the Society.

The officers have visited nearly all of the Chapters of the Society during the past year. The Committee on Constitution and By-Laws has been very active in having a new Charter prepared and in drafting several By-Law Changes which are being submitted to the Society.

The Meetings Committee has prepared a program for the Annual Meeting. At the October Meeting a decision was reached to hold the 7th International Heating and Ventilating Exposition in January 1947, in connection with the 53rd Annual Meeting of the Society, and annually thereafter as long as desirable.

The budget for the fiscal period November 1, 1945 to October 31, 1946 was adopted providing for an estimated income of $117,285.00 and expenditure of $115,405.00.

The award of the F. Paul Anderson Medal was authorized and arrangements were made for its presentation at the 52nd Annual Meeting.

Respectfully submitted,

THE COUNCIL.

REPORT OF SECRETARY

In the 12-month period between the 51st Annual Meeting in Boston and the 52nd Annual Meeting in New York there have been many problems for the Secretary's office to meet. One of the most vital was the matter of personnel which was only partially solved. This accounts for some of the delay in acknowledging receipt of dues and the preparation of TRANSACTIONS.

With an increase in membership of over 700 the slogan of the Membership Committee of 4500 in '45 became a reality. The Admission and Advancement Committee held frequent meetings both in New York and elsewhere to do its work which was exceeded in volume only once before in Society history. New membership literature has been prepared and is being sent to the list of prospects compiled by Chapter Membership Committees.

An important new duty was to handle the details of the assignment of speakers to chapters under the auspices of the Chapter Relations Committee. The trips to chapters for the officers were arranged.

Assistance has been given to many members who desired new connections after leaving the service. There are still 316 in service, the top figure being 427 earlier this year.

Publication matters have taken a great many hours in preparing the monthly JOURNAL, completion and production of THE GUIDE, membership list and TRANSACTIONS. It is pleasant to report that the 50th Anniversary edition of the TRANSACTIONS went into the mail early in January and THE GUIDE 1946 is on the press and will be ready for distribution in March. Members will find a much larger text and more firms represented in the catalog data section this year.

Editions had to be held to a minimum as there were difficulties experienced in obtaining paper and the quality is not up to previous standards. Costs increased greatly and a new selling price for THE GUIDE will be effective this year.

An increase in the advertising rates has been authorized effective next year because of the wider distribution of THE GUIDE and its increase in size.

The Society now has 37 chapters and it is anticipated that one or more will be chartered soon. Activity in the Norfolk area indicates the organization of an Old Dominion Chapter this year.

All of the routine actions required by the Constitution and By-Laws have been taken with regard to membership, finances, publications and other activities and the reports of the various committees will indicate the extent of the work accomplished.

During this year the Society has suffered great losses in the deaths of its Vice President, W. A. Russell, three of its past presidents, Thornton Lewis, D. S. Boyden,

C. B. J. Snyder, a former treasurer, F. D. Mensing, its Research Laboratory Director Comdr. Ferry C. Houghten and 20 other members.

The known war toll has been five members out of over 400 members in service.

Correspondence is now opening up with a great many of our foreign members who have been isolated from the United States during the war.

It is indeed a pleasure for me to tell you that the many activities of the Society could not have been carried on without the full cooperation and loyalty of the staff.

This year was one of the most successful in the history of the Society and all who have participated, the officers, council, committees, and chapter officers can find satisfaction in the prediction made at the 50th Annual Meeting, that the Society was looking forward to a greater future.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

Accountants' Report

TUSA & LABELLA

CERTIFIED PUBLIC ACCOUNTANTS
52 WILLIAM STREET
NEW YORK

AMERICAN SOCIETY OF HEATING AND
VENTILATING ENGINEERS,
51 Madison Avenue,
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and the related funds for the fiscal year ended October 31, 1945, and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1945. Also, for the fiscal year then ended, we traced the recorded cash receipts into the depositories; we inspected the cancelled bank checks which we compared with the record of cash disbursements; we supported the disbursements by payment vouchers; we accounted for the dues income and interest income from savings accounts and securities.

A Balance Sheet reflecting the financial condition of the Society as of the close of business, October 31, 1945 is submitted herewith and your attention is directed to the following comments thereon:

CASH

The Cash on Deposit was verified by direct communication with the commercial and savings banks listed in the schedule of cash included in this report and reconcilement of the amounts reported to us with the respective balances reflected by the books of the Society.

Checks representing the Cash on Hand for Deposit were inspected by us and the Petty Cash counted.

MARKETABLE SECURITIES

The securities, shown on the subjoined schedule, were verified by direct communication with the Bankers Trust Company, where same are deposited for safe-keeping. This Asset has been included in the Balance Sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

CERTIFICATE OF INDEBTEDNESS

The Certificate of Indebtedness issued to the Society was verified by inspection of the instrument.

The installments and interest on this indebtedness are being paid as due.

ACCOUNTS RECEIVABLE

A trial balance of the membership dues receivable taken as of the close of business October 31, 1945 was classified as to membership and aged as follows:

CLASSIFICATION	Amount
Members	$2,189.00
Associates	1,558.00
Juniors	173.50
Students	3.00
Total	$3,923.50

AGING	Amount
Dues Invoiced during 1945	$3,382.50
Dues Invoiced during 1944	156.50
Dues Invoiced prior years	384.50
Total	$3,923.50

The miscellaneous accounts receivable also were classified and aged as follows:

CLASSIFICATION	Amount
Guides (Advertising, Copy Sales)	$1,147.45
Transactions	66.50
Books and Reprints	129.54
Emblems	7.10
Sundry	52.13
Total	$1,402.72

AGING	Amount
Charges made during October, 1945	$ 205.19
Charges made during September	166.25
Charges made during August, 1945	44.41
Charges made Prior	986.87
Total	$1,402.72

The Reserve for doubtful Dues and Miscellaneous accounts receivable, in our opinion, are ample to cover probable losses that might result during realization.

INVENTORIES

The emblems and TRANSACTIONS on hand on October 31, 1945, were counted by us. The Guide paper was verified by direct communication with the printers. All inventories were priced and computed by us. The TRANSACTIONS follow:

Volume	Year	Quantity	Price	Amount
Prior	Prior	1350	$1.00	$1,350.00
45	1939	111	1.66	184.26
46	1940	50	1.25	62.50
47	1941	19	1.32	25.08
47	1941	150 (unbound)	.96	144.00
48	1942	52	1.42	73.84
49	1943	307	1.39	426.73
			Total	$2,266.41

PERMANENT ASSETS

Furniture, Fixtures and Library are shown herein at the book values without appraisal by us, we did, however, provide for depreciation of furniture and fixtures at the rate of ten per cent per annum.

DEFERRED CHARGES

We have deferred to future income one-sixth of all the subscriptions paid to *H.P.A.C.* since the payment of same are on a calendar year basis and the fiscal year of the Society ends on October 31st.

ACCOUNTS PAYABLE

All purchase invoices found on file that were applicable to the operations of the current fiscal period were listed and the proper liability therefor reflected in the attached balance sheet.

TAXES WITHHELD

The sum of $497.30 represents Federal Income taxes withheld from salaries paid to employees during the month of October, 1945.

DEFERRED INCOME

On October 31, 1945, the prepaid dues by elected members were:

Members	$243.46
Associates	322.84
Juniors	63.50
Total	$ 629.80

In addition there were dues in the sum of $395.00 prepaid by members proposed for membership.

RESERVE FOR PUBLICATIONS

In accordance with provisions made both in the 1944 and 1945 budgets, we have reserved the sums of $3,500.00 and $4,500.00, respectively, to cover the publication of TRANSACTIONS Vols. 50 and 51.

FUNDS

An analysis of the following funds reflecting the changes that occurred in these accounts during the fiscal year ended October 31, 1945, is included herein:

General Fund
Reserve Fund
Endowment Fund
Permanent Building Fund
F. Paul Anderson Fund

On October 16, 1944, the Council adopted the following resolutions:

"That in accordance with Article B-XI, Section 8 of the By-Laws, that the Reserve Fund shall be increased to an amount equivalent to $15.00 per member as of October 31, 1944, by transfer of money in the General Fund to the Reserve Fund."

"That the amount of Initiation Fees in excess of the sum required to meet the requirements of the By-Laws for the Reserve Fund as of October 31, 1944, be allocated to the Permanent Building Fund."

We have allocated all the initiation fees collected during the current fiscal year to the Reserve Fund, since the membership at $15.00 per member totaled $65,910.00 and exceeds the Reserve Fund of $64,493.66 on October 31, 1945. For this reason no allocation of initiation fees was made to the Permanent Building Fund.

There is submitted herewith a complete financial report as prepared for the Committee on Research setting forth the financial condition of the Research Fund as of the close of business on October 31, 1945, and the results from operations for the fiscal year then ended.

INCOME AND EXPENSES

A statement in summary form reflecting the Income and Expenses and comparable items budgeted for the fiscal year ended October 31, 1945, of the Society, THE GUIDE and Research follows:

SOCIETY	Actual	Budgeted
Income	$67,604.84	$61,785.00
Expenses	67,115.72	63,905.00
NET INCOME OR OUT-GO	489.12	2,120.00

GUIDE		
Income	61,242.22	54,000.00
Expenses	42,764.19	37,800.00
NET INCOME	18,478.03	16,200.00
RESEARCH		
Income	54,062.94	86,800.00
Expenses	51,676.76	76,550.00
NET INCOME OR OUT-GO	2,386.18	10,250.00
TOTAL NET INCOME	$21,353.33	$24,330.00
CAPITAL ITEMS		
New Equipment for Research	$ 5,862.42	$10,000.00
Initiation Fees to Reserve Fund	$ 7,091.00	$ 3,700.00

Respectfully submitted,

TUSA & LABELLA (*Signed*),
CERTIFIED PUBLIC ACCOUNTANTS.

Dated December 14, 1945.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.
October 31, 1945

ASSETS

GENERAL FUND			
CASH			
On Deposit		$33,690.13	
On Hand		100.00	$33,790.13
INVESTMENTS (AT COST)			
Securities (Market Value $23,665.00)		22,900.00	
Add: Accumulated Interest	$1,290.00		
Add: Accrued Interest	105.00	1,395.00	24,295.00
CERTIFICATE OF INDEBTEDNESS			75.52
ACCOUNTS RECEIVABLE			
Membership Dues		3,923.50	
Less: 40% for Research	2,110.33		
Less: Reserve for Doubtful	290.63	2,400.96	
		1,522.54	
Advertisers and Sundry Debtors	1,402.72		
Less: Reserve for Doubtful	112.50	1,290.22	2,812.76
INVENTORIES			
TRANSACTIONS		2,266.41	
Emblems		425.41	
GUIDE Paper		2,394.49	5,086.31
EXCHANGE			61.00

BALANCE SHEET

ASSETS

(continued)

PREPAID TRAVELING			67.44	
PERMANENT				
Library		300.00		
Furniture and Fixtures	3,248.44			
Less: Reserve for Depreciation	2,067.25	1,181.19	1,481.19	
DEFERRED CHARGES				
Prepaid *H.P.A.C.* Subscriptions			1,320.28	$ 68,989.63
RESERVE FUND				
CASH				
On Deposit		3,921.10		
On Hand for Deposit		7,181.00	11,102.10	
Securities at Cost (Market Value $53,394.75)		50,719.06		
Add: Accumulated Interest	2,638.75			
Add: Accrued Interest	33.75	2,672.50	53,391.56	64,493.66
ENDOWMENT FUND				
CASH				
On Deposit		2,660.51		
On Hand for Deposit		212.50	2,873.01	
Securities at Cost (Market Value $23,781.45)		22,005.65		
Add: Accumulated Interest	1,195.85			
Add: Accrued Interest	54.69	1,250.54	23,256.19	26,129.20
F. PAUL ANDERSON FUND				
CASH				
On Deposit		8.98		
On Hand for Deposit		12.50	21.48	
Securities at Cost (Market Value $988.00)		1,000.00		
Add: Accrued Interest		12.50	1,012.50	1,033.98
PERMANENT BUILDING FUND				
CASH				
On Deposit		5,889.23		
On Hand for Deposit		684.92	6,574.15	
Due from Reserve Fund			1,551.61	8,125.76
				$168,772.23

LIABILITIES AND FUNDS

GENERAL FUND

LIABILITIES

ACCOUNTS PAYABLE			$ 886.05	
FEDERAL WITHHOLDING TAXES			497.30	
ACCRUED ACCOUNTS				
Additional Compensation to Employees			4,449.14	
DEFERRED INCOME				
Prepaid Membership Dues	$629.80			
Less: 40% Prepaid to Research	226.52	$ 403.28		
Dues Prepaid by Candidates for Membership		395.00		
Fees Prepaid by Servicemen		68.50	866.78	
RESERVE FOR PUBLICATIONS				
TRANSACTIONS (1944), Vol. 50			3,431.89	
TRANSACTIONS (1945), Vol. 51			4,500.00	
TOTAL LIABILITIES			14,631.16	
GENERAL FUND			54,358.47	$ 68,989.63

Note "A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

RESERVE FUND

Due to Permanent Building Fund		$ 1,551.61	
Principal	$61,946.00		
Unexpended Income	996.05	62,942.05	64,493.66

ENDOWMENT FUND

Principal	25,378.40	
Unexpended Income	750.80	26,129.20

F. PAUL ANDERSON FUND

Principal	1,021.48	
Unexpended Income	12.50	1,033.98

PERMANENT BUILDING FUND

Principal	8,036.86	
Unexpended Income	88.90	8,125.76
		$168,772.23

STATEMENT OF INCOME AND EXPENSES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(For the Fiscal Year Ended October 31, 1945)

INCOME

INCOME FROM MEMBERS				
DUES—RENEWALS				
Members and Associates	$56,177.65			
Less: Cancellations	458.50	$55,719.15		
Less: 40% to Research Fund		22,287.66	$33,431.49	
Juniors and Students		1,355.50		
Less: Cancellations		16.50	1,399.00	$ 34,770.49
DUES—NEW MEMBERS				
Members and Associates		7,425.00		
Less: 40% to Research Fund		2,970.00	4,455.00	
Juniors and Students			365.50	4,820.50
TOTAL DUES				39,590.99
OTHER INCOME				
Initiation Fees			7,091.00	
Emblems and Certificate Frames			786.52	7,877.52
TOTAL INCOME FROM MEMBERS				47,468.51
INCOME FROM PUBLICATIONS				
Editorial Contract			$17,249.96	
GUIDE Sales and Advertisements—				
Per Schedule Attached			60,797.22	
Sale of TRANSACTIONS			1,560.63	
Income from Books, Reprints, Etc.			188.82	
Servicemen's Fees			1,014.50	80,811.13
INCOME FROM INVESTMENTS				
Interest—Savings Accounts			85.60	
Interest—Securities			480.00	
Interest—Certificate Frames			1.82	567.42
TOTAL INCOME				$128,847.06

EXPENSES

OPERATING EXPENSES		
President's Fund	$ 1,462.89	
Council Travel and Meetings	1,348.40	
Membership Committee	1,063.50	
Admissions and Advancement Committee	421.48	

EXPENSES

Operating Expenses (*continued*)

Constitution and By-Laws Committee	207.11		
Nominating Committee	24.94		
Cardex Books to Chapters	154.76		
Chapter Relations Committee and Records	229.85		
Executive Committee Allowance	13.43		
Finance Committee	24.27		
Chapter Delegates Travel	3,745.26		
A. S. A. Membership	100.00		
Membership Certificates	1,069.49		
Medals and Awards	70.00	$ 9,935.38	
MEETING EXPENSES			
Meetings	1,273.27		
Speakers to Chapters	1,253.99		
Chapter Meeting Allowance	650.00	3,177.26	
PUBLICATION EXPENSES			
Members' Subscriptions, *H.P.A.C.*	7,667.07		
GUIDE Publication and Distribution—Per Schedule Attached	42,764.19		
TRANSACTIONS	5,190.04		
Membership Roll	738.30		
Standards (Including Codes)	5.46		
Publication Committee Expense	97.55	56,462.61	
HEADQUARTERS' EXPENSES			
Salaries—Secretary and Staff	25,762.48		
Additional Compensation	4,449.14		
Traveling—Secretary and Staff	1,030.47		
Rent and Light	3,803.82		
Telephone	998.58		
Telegraph	333.10		
Postage	2,642.10		
General Printing	650.71		
Office Supplies	894.55		
Addressing and Address Changes	221.79		
Professional Services	950.00		
Bank Charges	82.21		
Depreciation of Furniture and Fixtures	324.84		
General Office Expense	1,148.58		
TOTAL HEADQUARTERS' EXPENSES	43,292.37		
Less: 30% Applicable to GUIDE	12,987.71	30,304.66	99,879.91
NET INCOME			28,967.15
DEDUCT: COUNCIL APPROPRIATIONS			
Initiation Fees to Reserve Fund		7,091.00	
Special Appropriation to Research		10,000.00	17,091.00
EXCESS OF INCOME OVER EXPENSES FOR THE YEAR			$ 11,876.15

BUDGET COMPARISON—SOCIETY ACTIVITIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(For the Fiscal Year Ended October 31, 1945)

INCOME

	Actual	Budgeted	Increases	Decreases
MEMBERSHIP INCOME				
DUES—RENEWALS				
100—Members	$ 20,936.70	$ 21,600.00		$ 663.30
101—Associates	12,494.79	10,800.00	$ 1,694.79	
102—Juniors	1,336.00	1,400.00		64.00
103—Students	3.00	45.00		42.00
	34,770.49	33,845.00	1,694.79	769.30
DUES—NEW MEMBERS				
104—Members	1,836.00	2,700.00		864.00
105—Associates	2,619.00	1,080.00	1,539.00	
106—Juniors	347.50	400.00		52.50
107—Students	18.00	30.00		12.00
	4,820.50	4,210.00	1,539.00	928.50
TOTAL DUES INCOME	$ 39,590.99	$ 38,055.00	$ 3,233.79	$ 1,697.80
OTHER INCOME				
108—Initiation Fees	7,091.00	3,700.00	3,391.00	
109–110—Emblems and Certificate Frames	786.52	200.00	586.52	
	7,877.52	3,900.00	3,977.52	—0—
TOTAL INCOME FROM MEMBERS	47,468.51	41,955.00	7,211.31	1,697.80
INCOME FROM PUBLICATIONS				
115—Editorial Contract	17,249.96	17,250.00		.04
116—GUIDE Sales and Advertisements	60,797.22	54,000.00	6,797.22	
117—Sale of TRANSACTIONS	1,560.63	700.00	860.63	
118—Income from Books, Reprints, etc.	188.82	400.00		211.18
119—Sales Codes	—0—	25.00		25.00
120—Servicemen's Fees	1,014.50	450.00	564.50	
	80,811.13	72,825.00	8,222.35	236.22

INCOME FROM INVESTMENTS				
125—Interest — Savings Accounts	85.60	250.00		164.40
126—Interest—Securities	480.00	750.00		270.00
127—Interest—Cert. of Indebt.	1.82	5.00		3.18
	567.42	1,005.00		437.58
COLLECTION OF PRIOR YEAR'S DUES	2,196.50	1,500.00	696.50	—0—
TOTAL INCOME	$131,043.56	$117,285.00	$16,130.16	$ 2,371.60

EXPENSES

OPERATING EXPENSES				
150—President's Fund	$ 1,462.89	$ 2,500.00		$ 1,037.11
151—Council Travel to Meetings	1,348.40	2,500.00		1,151.60
160—Executive Committee	13.43	100.00		86.57
161—Finance Committee	24.27	200.00		175.73
162—Membership Committee	1,063.50	2,500.00		1,436.50
170—Admissions and Advancement Committee	421.48	500.00		78.52
171—Constitution and By-Laws	207.11	150.00	$ 57.11	
172—Nominating Committee	24.94	150.00		125.06
173D—Cardex Books to Chapters	154.76		154.76	
173B—Chapter Delegates Travel	3,745.26	3,300.00	445.26	
173C—Chapter Relations Committee	229.85	350.00		120.15
174—War Service Committee		1,000.00		1,000.00
201—*A.S.A.* Membership	100.00	100.00		
204—Membership Certificates	1,069.49	200.00	869.49	
206—Medals and Awards	70.00	200.00		130.00
	9,935.38	13,750.00	1,526.62	5,341.24
MEETING EXPENSES				
163—Meetings	1,273.27	2,500.00		1,226.73
173A—Speakers to Chapters	1,253.99	1,500.00		246.01
327—Chapter Meeting Allowance	650.00	900.00		250.00
	3,177.26	4,900.00	—0—	1,722.74
PUBLICATION EXPENSES				
200—Members' Subscriptions, *H.P.A.C.*	7,667.07	7,500.00	167.07	
305-326—GUIDE Publication and Distribution	42,764.19	37,800.00	4,964.19	
202—TRANSACTIONS	5,190.04	4,500.00	690.04	
203—Membership Roll	738.30	1,500.00		761.70
164—Standards (including Codes)	5.46	150.00		144.54
207—Publication Committee Expenses	97.55		97.55	
	56,462.61	51,450.00	5,918.85	906.24

HEADQUARTERS' EXPENSES				
210—Salaries — Secretary and Staff	25,762.48	30,000.00		4,237.52
211—Additional Compensation	4,449.14	3,200.00	1,249.14	
212—Traveling—Secretary and Staff	1,030.47	1,500.00		469.53
213—Rent and Light	3,803.82	4,000.00		196.18
214—Telephone	998.58	700.00	298.58	
215—Telegraph	333.10	200.00	133.10	
216—Postage	2,642.10	1,800.00	842.10	
217—General Printing	650.71	1,000.00		349.29
218—Office Supplies	894.55	600.00	294.55	
219—Addressing and Address Changes	221.79	250.00		28.21
220—Professional Services	950.00	800.00	150.00	
221—Bank Charges	82.21	100.00		17.79
222—Depreciation—Furniture and Fixtures	324.84	250.00	74.84	
223—General Office Expense	1,148.58	750.00	398.58	
TOTAL HEADQUARTERS' EXPENSES	43,292.37	45,150.00	3,440.89	5,298.52
Less: 30% Charge to GUIDE	12,987.71	13,545.00	—0—	557.29
	30,304.66	31,605.00	3,440.89	4,741.23
INITIATION FEES TO RESERVE FUND	7,091.00	3,700.00	3,391.00	—0—
COUNCIL APPROPRIATION TO RESEARCH	10,000.00	10,000.00	—0—	—0—
TOTAL EXPENSES	$116,970.91	$115,405.00	$14,277.36	$12,711.45

RESEARCH BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.
October 31, 1945

ASSETS

RESEARCH FUND		
CASH ON DEPOSIT		
Treasurer's Account—Bankers Trust	$ 3,200.59	
Secretary's Account—Chase National Bank	956.72	
Director's Account—Cleveland Trust Co.	769.09	
Thrift Account—Bank for Savings	3,426.90	$ 8,353.30
ON HAND FOR DEPOSIT		
Treasurer's Account—Bankers Trust Co.	4,945.15	
Director's Account—Cleveland Trust Co.	1,610.08	
Secretary's Account—Chase National Bank	2,921.62	9,476.85
ESCROW FUNDS IN TRANSIT		
To the Cleveland Trust Co.		8,000.00

ASSETS (*continued*)

RESEARCH FUND			
ON HAND			
Cleveland, Ohio		50.00	$25,880.15
TRAVEL IMPREST FUND			200.00
PERMANENT			
Laboratory Equipment, Furniture and Fixtures	13,466.82		
Less: Reserve for Depreciation	1,580.08	11,886.74	
Tools and Supplies		300.00	12,186.74
DEFERRED CHARGES			
Prepaid Rent		1,325.00	
Withholding Taxes		63.20	1,388.20
TOTAL RESEARCH FUND			39,655.09
RESEARCH ENDOWMENT FUND			
CASH ON DEPOSIT			
Bank for Savings			596.91
			$40,252.00

LIABILITIES AND FUNDS

RESERVE FUND			
ACCOUNTS PAYABLE		$ 250.00	
DEFERRED INCOME			
PROJECT			
Summer Cooling	$1,859.45		
Air Friction in Ducts and Fittings	2,016.46		
Shock Effect in Summer Air Conditioning	66.50		
Weather Data for Design	600.00		
Heat Losses Due to Infiltration	188.75		
Solar Heat Gain of Insulated Walls	75.00		
Field Studies of Radiant and Panel Heating	91.66		
Gravity Circulation of Water	25.00	4,922.82	
DUES—MEMBERS AND ASSOCIATES			
40% of 1946 Dues Prepaid in 1945		226.52	$ 5,399.34
RESEARCH FUND			34,255.75
TOTAL RESEARCH LIABILITIES AND FUND			$39,655.09
RESEARCH ENDOWMENT FUND			
Principal			596.91
			$40,252.00

NOTE "A"—There exists a contingent asset in the sum of $2,110.33 arising from 40% of the Society's members and associates dues pending collection by the Society.

NOTE "B"—This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

RESEARCH BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(For the Fiscal Year Ended October 31, 1945)

INCOME

	Actual	Budget Provision	Increases	Decreases
A.S.H.V.E.—DUES	$24,630.65	$25,120.00		$ 489.35
CONTRIBUTIONS				
Navy Contract	1,600.00	1,600.00		
Earmarked	9,419.88	15,000.00		5,580.12
General	8,361.59	35,000.00		26,638.41
Special Allocation from Council	10,000.00	10,000.00		
Interest	50.82	80.00		29.18
TOTALS	$54,062.94	$86,800.00	—0—	$32,737.06

EXPENSES

CHAIRMAN AND COMMITTEE				
Traveling	$ 608.02	$ 1,000.00		$ 391.98
Chairman's Office	340.45	600.00		259.55
Professional Services	250.00	200.00	$ 50.00	
RESEARCH LABORATORY				
Salaries — Administrative and Technical	21,490.52	41,000.00		19,509.48
Salaries—Clerical	5,088.60	5,200.00		111.40
Traveling	1,862.41	2,500.00		637.59
Postage	362.51	350.00	12.51	
Telephone and Telegraph	621.30	350.00	271.30	
Office Expenses and Supplies	585.03	500.00	85.03	
Printing and Multigraphing	572.72	600.00		27.28
Library and Periodicals	255.99	300.00		44.01
Laboratory Materials and Supplies	3,465.83	2,500.00	965.83	
Insurance	—0—	300.00		300.00
Depreciation—Equipment	1,239.37	750.00	489.37	
Unallocated	337.77	1,500.00		1,162.23
Taxes	2,180.00	2,000.00	180.00	
Property Insurance	168.40	200.00		31.60
Water, Gas, Heat and Electricity	1,109.71	1,300.00		190.29
Janitor Services and Supplies	1,274.09	1,000.00	274.09	
Building Maintenance	83.70	100.00		16.30
COOPERATIVE RESEARCH	8,248.86	12,700.00		4,451.14
NAVY CONTRACT	1,600.00	1,600.00		
SPECIAL COUNCIL APPROPRIATION	5,862.42	10,000.00		4,137.58
TOTALS	$57,607.70	$86,550.00	$ 2,328.13	$31,270.43

President Winslow then announced that the major business at the first session was the consideration of certain changes to be made in the Charter, Constitution and By-Laws.

The legal counsel, L. P. Eisner, prepared a revised Charter when it was found that the original charter under which the Society had operated for most of its 50 years, did not adequately cover the various activities now being carried on by the Society.

It was moved by R. H. Carpenter, New York, N. Y., chairman of the Committee on Constitution and By-Laws, and seconded by Past President Homer Addams, New York, that the new Charter be adopted. There was no discussion of the motion and President Winslow announced that the motion was unanimously carried and the proposed Charter was adopted.

Change in Purposes, Powers and Other Provisions and Increase in Number of Directors

OF

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

The number of directors previously authorized is twelve (12). The number of directors as increased by this certificate hereafter shall be not less than thirteen (13) nor more than twenty-one (21). The Board of Directors shall be styled "The Council."

The purposes, powers and other provisions to be eliminated are:

"That the purposes for which said corporation is organized are, the promotion of the arts and sciences connected with heating and ventilation, the improvement in the mechanical construction of heating and ventilating apparatus, the establishment of a minimum standard of heating and ventilation, the encouragement of legislation compelling the ventilation of buildings in accordance with the standard of the Society and of legislation favorable to improvement in the arts of heating and ventilation and the opposition of legislation inimical to the business of the engineer, the reading, discussion and publication of professional papers and the interchange of knowledge and experience among its members, the establishment of a uniform scale of prices for all professional services, and the encouragement of good fellowship among its members."

The purposes, powers and other provisions to be substituted for the foregoing are as follows:

(a) To advance, promote and cultivate the arts and sciences of heating, ventilating, cooling and air conditioning and the allied arts and sciences.

(b) To conduct, promote and coordinate fundamental and practical research, and to study and develop new principles, uses, products, inventions and methods in the fields of heating, ventilating, cooling and air conditioning and the allied arts and sciences.

(c) To cooperate with universities, colleges, schools and other organizations and groups, including governmental agencies, for the investigation of research subjects.

(d) To promote the advancement of professional knowledge, educational programs and standards for the instruction and technical training of persons engaged in the engineering profession and students and teachers of heating, ventilating, cooling and air conditioning and the allied arts and sciences.

(e) To promote the interchange of knowledge and opinions and to publish, diffuse and disseminate in any language and in every form and medium now or hereafter known, information concerning heating, ventilating, cooling and air conditioning and the fields related thereto.

(f) To hold meetings and arrange events and programs for the reading and discussion of papers and the exhibition and display of items in the fields of heating, ventilating, cooling and air conditioning and the allied arts and sciences.

(g) To produce, publish and foster, in any language and in every form and medium, publications, books, periodicals, programs, data, reports, standards, guides, educational programs and codes relating to heating, ventilating, cooling and air conditioning and the allied arts and sciences.

(h) To secure copyright registrations, Letters Patent, or equivalent protection in the name of the corporation or others, in all countries of the world, of patentable inventions and discoveries and copyrightable writings and material, and to license the same and dispose thereof, in whole or in part, subject to the following provisos: (1) That the foregoing activities shall not be conducted with the view of pecuniary profit to the corporation or to any member, officer, director, agent or employee thereof; (2) that the surplus proceeds, if any, derived by the corporation from the foregoing activities shall be used only in furtherance of the corporation's objects and purposes and in the performance of its public obligations; and (3) that if a patentable invention or discovery be made or a copyrightable writing produced in cooperation with the United States Government, that the same may, at the option of the directors, be dedicated to the people or assigned, without consideration, in whole or in part, to the Government of the United States.

(i) To maintain a respresentative and centralized agency to consider and act upon matters affecting the design, construction, research, development and study of heating, ventilating, cooling and air conditioning and the allied arts and sciences.

(j) To maintain a centralized agency where engineers and persons engaged in the fields of heating, ventilating, cooling and air conditioning and the allied arts and sciences, may register and file biographical and other data for the confidential information of prospective employers.

(k) To encourage the establishment of and compliance with codes of ethics, codes and rules of practice, and fair standards of professional conduct; to encourage uniformity and certainty in customs and usages among those engaged in the practice of engineering and the arts and sciences of heating, ventilating, cooling and air conditioning and the industries related thereto; to cooperate with governmental agencies in the preparation of rules and standards for the licensing of professional engineers; and to notify agencies having disciplinary jurisdiction of unworthy and unauthorized practices.

(l) To develop good will and cordial relations among its members and persons practicing or teaching or studying the arts and sciences of heating, ventilating, cooling and air conditioning and allied arts and sciences and to foster among said members and persons a mutuality of interest and understanding.

(m) To assist in the formation of local chapters and student and other branches, and to regulate, operate and control the same under the direction and at the pleasure of the corporation, but no local chapter or branch shall subject the corporation to any financial or other obligation except such as the corporation may voluntarily assume.

(n) To receive, acquire, hold and maintain any property, real or personal, without limitation as to amount or value, for any of the corporation's objects, by way of bequest, devise, gift, purchase or lease, to invest and re-invest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest or principal.

(o) To cooperate with and contribute to universities, colleges, schools and other organizations and groups having the same or similar objects; to establish scholarships and make awards in recognition of notable scientific achievement or outstanding services performed in the fields of heating, ventilating, cooling and air conditioning or the allied arts and sciences.

(p) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(q) No substantial part of the activities of this corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation.

(r) The territory in which the corporation's operations are principally to be conducted is in all parts of the United States and its possessions and in the Dominion of Canada, but the corporation may on occasions extend its operations to other parts of the world.

(s) No officer, director or member of this corporation shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in affecting one or more of its purposes as the Board of Directors may determine.

(t) Each director of the corporation shall be indemnified by the corporation against expenses actually and necessarily incurred by him in connection with the defense of any action, suit or proceeding in which he is made a party by reason of his being or having been a director of the corporation, except in relation to matters as to which he shall be adjudged in such action, suit or proceeding to be liable for wilful negligence, misfeasance or misconduct in the performance of his duties as director; such right of indemnification shall not be deemed exclusive of any other right to which he may be entitled under any by-law, agreement, vote or otherwise.

President Winslow then referred to the proposed amendment to **Article B-VIII,** *Section 11* of the By-Laws, which had been prepared in accordance

with action taken at the 51st Annual Meeting in Boston, Mass., and refers to a change in the number of members and the method of electing members of the Nominating Committee.,

Mr. Carpenter read the resolution adopted at the 51st Annual Meeting at Boston, which was as follows:

THAT the resolution be referred to the Council for transmission to the Constitution and By-Laws Committee with instructions to present the resolution in proper form for Society action at a later meeting.

Mr. Carpenter announced that a copy of the proposed amendment had been mailed to each member of the Society under date of December 15, 1945, as required by the By-Laws.

It was moved by Mr. Carpenter and seconded by E. M. Mittendorff, Chicago, that the proposed amendment to **Article B-VIII,** *Section 11* of the By-Laws be adopted.

President Winslow declared that the motion was now ready for discussion.

DISCUSSION

F. W. LEGLER, Minneapolis, Minn.: The Minnesota chapter has discussed this proposed change of the Constitution. At the Annual Meeting in Boston, when this was brought up, there was considerable opposition to it. It was brought up on a Monday when there were not very many members present; instead of Wednesday as new business, and was voted on to be presented as it has been today.

I am opposed to the entire procedure for this reason: out of the 11 members of the Nominating Committee, four are to be appointed by the Council. That means 40 per cent of the Committee which is to propose names for the governing body of our organization, are to be appointed by the governing body. I am not opposed to reducing the number of members of the Nominating Committee from 37 to 11, but I am opposed to having four of the 11 appointed. They should not be appointed by the Council but should be elected by the membership.

B. L. EVANS, St. Louis, Mo.: We are opposed to this amendment for the same reason that Mr. Legler has just outlined, plus one additional reason. We do not believe it is possible for a representative of a region to determine which chapter members might be qualified to be Council members or national officers.

H. R. ALLONIER, Columbus, Ohio: In speaking for the Central Ohio Chapter, I am bringing you not my opinion but the opinion of the chapter's members.

Two years ago Mr. Blankin did a very fine job, and we have no feeling toward him or his proposition other than that we do not feel that the few men sitting here should decide on an issue involving some 4300 members. The small percentage of men present, some of whom probably are not even delegates or members and may not vote, should not decide an issue that involves the entire membership. It is felt that if this is such a good thing it should be voted upon by letter and give each member permission to vote on this change in the Nominating Committee.

It has been said that the Nominating Committee of 37 members is unwieldly. Well, I happened to be on the Nominating Committee last year. We started at 12:30 p.m., had a very fine lunch, and by 5:30 p.m. we had nominated whom we think, and believe you will agree, are some very fine men.

If a Nominating Committee of 37 members is unwieldy, then will not 37 delegates when they meet the first day this next January be just as unwieldy in trying to pick out the delegates for the Nominating Committee?

PRESIDENT WINSLOW: I would like to say that we have just been looking up the Constitution. This is the only machinery provided for amending the By-Laws. Amendments to the Constitution can be adopted by letter ballot but the only way that the By-Laws can be amended, as distinct from the Constitution, is in this manner, as now proposed, by vote at a meeting of the Society.

F. E. P. KLAGES, Greensboro, N. C.: I have been asked by a majority of our chapter to go on record as supporting the position of the representative from the Indiana Chapter.

M. W. BISHOP, Milwaukee, Wis.: I am not speaking officially for the Wisconsin Chapter, but merely as an individual who has had considerable experience on nominating committees for quite some years. Basically I think that reduction in the size of the Nominating Committee is a good thing. I further think that the plan set forth here comes closer perhaps to providing the type of procedure that we want or that can be acceptable to all members than anything we have had before.

However, the geographical distribution of this plan is, in my opinion, entirely out of line. Adding up either the chapters or the individual members that are involved in this procedure you will find: *Zone 1* has two chapters and 134 members. They have one vote. *Zone 2,* six chapters and 708 members; one vote. *Zone 3,* eight chapters and 751 members; one vote. Beyond that point you have *Zone 4,* with about 400 members; *Zone 5,* 427; *Zone 6,* 273, and *Zone 7,* the Canadian Zone, 253.

The Committee by its efforts has brought this thing out in very workable form but states that it would have been very difficult to get a more equitable distribution. We realize it is a difficult job, however, I would make one suggestion. If, for example, the State of New York, was moved from *Zone 2,* you would then have a distribution something like this: *Zone 1,* five chapters, 488 members; *Zone 2,* six chapters, 594 members, *Zone 3,* five chapters, 511 members. Then the remaining zones would be left as is. This would give a far more representative distribution of membership and a much more equal voice in the nomination of our officers.

As I stated before, I think that Mr. Carpenter's committee has done a good job in bringing this thing down to a reasonably acceptable point but I cannot support it as it now stands; with 135 members in one zone who have an equal vote with 754 in another and 705 in another. I believe that a more equitable distribution could be affected and that if this was done, there might be more general support of this proposal.

PRESIDENT WINSLOW: Is that made as a motion to amend?

MR. BISHOP: No. It is made merely as an individual suggestion. I do not know whether enough other members would agree with me to make it worth while as a motion. Let the discussion go on further, and see how others may feel about it.

PRESIDENT WINSLOW: I am advised by counsel that any amendment which changes the fundamental intent of this program would not be in order but on the other hand a definition of zones would be modification of detail and therefore permissible. At present the motion is before you for the adoption of this change as it stands. Is there any further discussion?

M. F. BLANKIN, Philadelphia, Pa.: I feel I am speaking not only for myself but also for the large group of members of this Society, who are not members of any chapter. The inclusion of four members of the committee to be elected by the Council, was decided in Boston for the purpose of giving representation to those members who do not belong to a chapter. I think this is fair, quite democratic, and quite representative. I do not know the current percentage of non-chapter members compared with chapter members but the proportion was about four to seven.

I am glad that Mr. Bishop brought out the wide difference between the various zones. I do not like the idea of New England with 134 members being represented by two chapters but I was willing to accept it, since it could be changed later. Perhaps the amendment suggested here is an excellent one. I am sure, as ruled by the chair that it can be determined from the floor because the original resolution stated, *Regions as determined* . . . so that it can be determined here on the floor of the meeting.

This matter has come up before and it is going to keep coming up until something is done regarding a change. We have 37 chapters today and I believe we have under consideration three new chapters. We will get more chapters. We will soon have 50 on the committee. We will have 60 on the committee. Some change will have to be made in the future. That was my sole reason for suggesting it last winter at Boston.

L. T. AVERY, Cleveland, Ohio: I served on the Constitution and By-Laws Committee with Mr. Carpenter when we presented our humble efforts at Cincinnati. If you will recall, we had thought to interpret by some mail return to a questionnaire what the consensus of the membership was, and it failed. At that time we recognized that a mail ballot or a mail survey of this organization hits too many busy people and may not be conclusive.

The By-Laws are very clear on how they are to be amended. The amendment to the By-Laws is handled as we are handling it here, by means of discussion rather than an open-and-shut yes-or-no written vote. It seems wise to have such a method of amending the By-Laws. If you have criticism of that method you have to go back and change the Constitution. That requires another job with the Constitution and By-Laws Committee.

I would point out that the new charter provides that the Council shall be not less than 13 nor more than 21. The nominating procedure is that the Council shall elect four members of the Nominating Committee, and neither those four members nor any of the seven members elected by the chapter delegates may be Council members. That answers the main question that we faced at Cincinnati as to appointing a self-perpetuating group. In other words, our procedure here is so democratic I doubt if you could think of any other combination which would represent, as Mr. Blankin stated, those who do not belong to chapters, and which would eliminate Council members themselves from being on the Nominating Committee.

At the same time this permits the chapter delegates to exercise a very real method of selection of seven of their own group, if they like, or outsiders, to go to the Semi-Annual meeting and be sure that they will be there to make the nominations.

MR. LEGLER: Why was this provision made that four of these proposed members of the Nominating Committee should be elected by the Council? Why cannot all members of the Nominating Committee be elected by the membership? That is my chief objection to this proposal as it stands now.

PRESIDENT WINSLOW: Because that was in the resolution adopted by the membership last year, and at the expense of being repetitious let me remind you again that this is the third time this matter has come up. At no time on any of those three occasions have these suggestions been made on the initiative of the Council. There has never been an attempt by the Council to get more power. Every time it has come up from the floor, it has been all up to you. The Council has taken no stand on this matter and has never initiated any of this legislation. It has merely put into form this proposal when ordered to do so. I do not care personally how you decide, but I hope the Society this afternoon will make up its mind and that if it votes against this proposal it will not be made again next winter.

R. H. CARPENTER, New York, N. Y.: I would just like to comment briefly in connection with the question that was raised by Mr. Legler relative to the four

members to be elected by the Council. Apparently he overlooked what Mr. Blankin just said: That in having those four members chosen by the Council, the fundamental idea was that they would look after the interests of the non-chapter members. These figures that I have here are not up to date as of the present but they were the last available figures when we attempted to make up these zones.

As of January 1, 1945, the total chapter membership was 2,905, and the total membership of the Society was approximately 4,300, which is roughly 35 and 65 per cent. I do not think anybody will dispute the statement that the present Nominating Committee represents only those 2,905 members who are chapter members. As far as democratic procedure is concerned, I trust you men have not overlooked the fact that our Chapter Delegates Committee, as it is now constituted, indicates that the Oklahoma chapter, with 11 members, has exactly the same representation as Philadelphia with 137 members, or Southern California with 160 members, or New York with 225 members.

MR. AVERY: I would like to ask Mr. Carpenter if there is a basis for geographical division into zones.

MR. CARPENTER: Not that I have been able to determine. In the explanatory letter that went out over the committee's signature with this proposed amendment, we tried to bring that out clearly. There were several schemes under discussion and the fourth one that we adopted seemed the best of the four. I must confess that I am intrigued with this suggestion to move New York into the New England group. I feel quite confident the other two members of the committee would have joined me in drawing our zones that way, if it had been suggested in time, because really that was a very troublesome thing. Speaking for myself only, I was not at all proud of the way we finally put it out but there did not seem to be anything else to do. I realized that the membership is very unevenly divided in those seven zones but at that time there seemed no preferable way to accomplish it.

W. A. DANIELSON, Memphis, Tenn.: With the growth of the Society it is only natural that if we are to function smoothly we shall have to break it up into geographical units. I have belonged to the *American Institute of Electrical Engineers* now for something over 30 years and that has been regionalized for some time. I think this is the first step in the right direction. In regionalizing you have two factors to consider: one is the number in an area and then the ability of those areas to work together.

I therefore move that in order to change this to a more rational basis, that New York be included in the New England region, and that West Virginia and Kentucky be included in the Mid-Western or Ohio region. This latter part I move because those groups naturally work together from West Virginia and from Kentucky with the Ohio region.

PRESIDENT WINSLOW: Is this amendment seconded? The chair hears no second.

J. S. LOCKE, Chicago, Ill.: I represent the Illinois Chapter. We have 164 members on this list and we are opposed to the change in spite of Oklahoma with 11. We have not had a vote of our chapter but we have discussed it at considerable length at our Board of Governors meetings and at regular chapter meetings. The consensus so far has been in favor of our present procedure.

We were particularly opposed to this election of four of the 11 members by the Council. It seemed to us that the men who are most active in the Society, are those who are also members of local chapters, and if non-chapter members wished to participate in a democratic election we would be very happy to have them join the Illinois chapter.

J. D. KROEKER, Portland, Ore.: The far western chapters have asked me to speak for them. We like this amendment. We think there are probably some features of

it, particularly the zoning, which may have to be changed later. That, however, is a detail. It was necessary to make compromises, and we feel that the compromises have been good. However, there is a collateral concern which the far western chapters have which possibly may not belong exactly in this discussion, but the chapters wished that it be brought out on the floor.

It concerns the representation at these Annual Meetings. We live a long way from here and the expense of traveling to the Annual Meetings is heavy. The chapters cannot bear it. So the Council very kindly has from year to year made provision for paying traveling expenses of chapter delegates. It is not included in the By-Laws or the Constitution.

We are somewhat concerned in this connection in that adoption of the amendment more or less as written, in which provision is made to pay the traveling expenses for the Nominating Committee, will eliminate the payment of expenses of the delegates to the Annual Meeting.

PRESIDENT WINSLOW: I might say that after the Charter is adopted the old Constitution will have to be replaced by a new one adopted under the Charter, and that will be a very good opportunity to make minor changes, but I do think it is very important at this Annual Meeting that the policy of the membership should be determined, whether they want one kind of Nominating Committee or the other. The details can be worked out in connection with the necessary revisions that will follow the Charter.

MR. BLANKIN: I would like to have the By-Laws read concerning the election of the chapter delegates, or how it is provided for in the Constitution. I thought Mr. Kroeker was wrong on that. It seems that was made a part of the By-Laws.

S. L. FURBER, Kansas City, Mo.: I would like to express the considered opinion of 102 of our members. We took the time at our last meeting to discuss this at length and I would like to say that we are unalterably opposed to it, particularly on the ground of the Council appointing four members, as has been frequently expressed here this afternoon. We feel, as Mr. Legler said, that the progress of this Society is largely in the hands of the chapter members. Are we to be an organization of chapters, or just loosely divided members? Frankly, I was alarmed that there were about 1400 members of the Society that are not members of the chapters. I think therein lies a fundamental issue: do we want to encourage or discourage that sort of thing?

PRESIDENT WINSLOW: The reference which Mr. Blankin mentioned is Article B-VIII *Section 14* which says:

Article B-VIII, *Section 14*. There shall be a standing committee known as the Chapter Delegates Committee. The membership of this Committee shall consist of one Delegate or Alternate from each local chapter, who shall be selected as each chapter may direct. The Chapter Delegates Committee shall meet at each Annual and Semi-Annual Meeting of the Society to discuss problems of mutual interest affecting the chapters. The Chapter Delegates shall elect a chairman to serve for a term of one year, from June to June. Meetings of the Chapter Delegates Committee may be attended by any local chapter officer, but each chapter shall have only one vote.

MR. BLANKIN: There is nothing said about expenses?

PRESIDENT WINSLOW: Not in that clause, no.

G. L. TUVE, Cleveland, Ohio: I do not represent anybody in particular but I have always championed the idea of geographical representation of the West. I am personally in favor of this proposal because I think it makes the West stronger, rather than weaker, in this organization. I also agree with the statements that have been made about the people who do not belong to chapters. Some of our outstanding members do not belong to chapters. I think they should be represented in some way.

Another item I would like to mention is that apparently as soon as a person starts giving several hundred hours a year to the Society as a member of the Council he

grows horns and a tail. I assure you that being a member of the Council is a very expensive business, if you do not have an expense account. Perhaps the objection to giving this nominating responsibility to the Council is based on the conviction that anybody who is dumb enough to devote so much of his time free, just is not responsible enough to be elected an officer!

Of course another indication on this whole matter is the fact that the Nominating Committee as it previously existed just got through nominating me for an office which in itself indicates that the system should be changed.

I would like to make one other remark. It seems that in this Society, it is very easy to stampede any business meeting (as has been done before) into turning thumbs down on anything that is proposed. I do not care particularly about this Nominating Committee proposal but I hope that this meeting will not get into the frame of mind of turning down everything that is proposed this afternoon.

H. J. RYAN, New York, N. Y.: As president of the New York chapter I would just like to add a word at this particular point about how we have talked about this thing; and mention what I think are perhaps common problems with most of you. We are in a rather thickly-populated area, and we are by actual members probably one of the largest chapters, with 225 members. We are not very proud of the fact that of the 450 members in this area of the National Society, we have in our chapter only about half of them, or a little more than half. We went into that about two years ago, when I was chairman of the membership committee, and tried to find out why. Results showed that the reason why those men were not active members of our chapter was because of the nature of their work, or for reasons other than personal ones, which were beyond our control in any way, and not because they would not like to be members. Those of you, coming from the West where you travel long distances, I am sure, can appreciate that it is not always possible to get all your members out to your chapter meetings, but out there perhaps you may feel that any man who is a member of a national organization should definitely belong to the group with which he is associated, as a local member of a chapter. I know one man said today that there are only five members of the Society in his territory who are not members of the local chapter. I think that is wonderful. I hope some day we shall have something like that in New York, but I cannot help but feel that in a thickly-populated area such as we have here in the East, and in other parts of the country, where people cannot get to chapter meetings we never shall be able to attain a full quota. Failing to attain that full quota we should not feel, that we, as chapter officers, have any right to speak for these men. If any of these men are here today, I think they should feel that it is their responsibility to talk to us who are chapter officers, and not leave it to us to make this decision year after year.

J. H. VAN ALSBURG, Holland, Mich.: During my Society activities I have been a member of four chapters. I think this is a little unusual because in addition, for a few years I did not belong to any chapter. I recently have been informed that I am again a member of my first chapter—namely, Western Michigan, which is 30 miles from my home. Over half of our Western Michigan members do not live ir a chapter meeting city.

By spending some time in four chapters, one soon gathers a cross-section of experience and opinion as to what members think of our governing body. It seems unusual, in view of what Professor Tuve has said, that most of the men who spend a lot of time gratis for the Society are looked upon as *suspicious characters.* I remember the day the President of my company said, "For goodness sake! Whom do you work for; the Research Committee or the company?" At one time I think almost 20 per cent of my time was spent in Society work. I have served on the Nominating Committee and we had to meet more than one afternoon. Maybe it was because

of my arguing; but I believe most Nominating Committees have deliberated more than a few hours.

Any time one has a large committee, the old saying is: *Send half of them on errands so the rest do some work.* If we are going to attempt to elect Council members, nominate them for election and we end up with 37 chapters represented on the committee, it is not a committee, it is a congregation!

Regardless of whether we are all in agreement on this suggested method, I believe we do agree that it has some merit. It does at least get us away from the congregation and back to a feasible committee!

Gentlemen, I propose the following: This is the third time the nominating method has been up for discussion. I am for most of it as suggested, and believe we cannot be much worse off. I would like to amend the present motion to the extent that the regional areas as defined by Mr. Bishop, namely, add New York to *Zone 1* and Ohio to *Zone 2.*

MR. BLANKIN: I second the amendment.

PRESIDENT WINSLOW: Will you discuss the amendment? Are you ready to vote on the amendment? Those in favor of the amendment say *Aye.* Contrary minded? It is so ordered, *the amendment is adopted.*

MR. VAN ALSBURG: We now have the main question before us. The suggested method for nominating is not all that we wish but it cannot be changed further here. Let us attempt something different and see if it is not a little more feasible than our present method.

I further suggest that we eliminate some of our discussion, which can be carried on all day, and either approve this motion and try it out or disapprove it. I am in favor of it.

PRESIDENT WINSLOW: I do not interpret that as a call for the question formally.

R. A. SHERMAN, Columbus, Ohio: Speaking individually and not for the Central Ohio Chapter, because Mr. Allonier has very ably presented the attitude of the chapter members. I am certainly for the plan that the nominating committee be composed of members representing the different regions. General Danielson has told you of the experience of the *American Institute of Electrical Engineers.* Others here, as well as I, could tell you of the experience of the *American Society of Mechanical Engineers,* who have some 17,000 or 18,000 members and for a number of years have had their nominating committee made up from eight regions. They have I believe, some 50 sections. Having served on the Nominating Committee with eight members I should hate to think of what it would be like if we had one man from each of the 50 sections. It is a big job in the *American Society of Mechanical Engineers* to nominate able men for office and I think it is just as big a job in this Society. I think that seven or 11 men could do a much better job than 35 or 36. But, this method of the Council electing four members, no matter how well intentioned it may be, always savors of packing by the *palace guard,* and from that standpoint I am definitely against it.

Mr. Blankin stated the reasons for this proposal which I can see are good reasons, but I should like to ask Mr. Carpenter and the members of the committee who worked so hard on this proposal if they did not consider the possibility of making it mandatory that these four members who are elected by Council be chosen from non-members of chapters. That I think could be very simply put in and I think it is in order to make a change like that.

I would move that in the clause where it says, *four members and one alternate shall be chosen by a majority of the Council,* these words be inserted: *from members who are not affiliated with chapters.*

MR. KROEKER: I second the amendment.

PRESIDENT WINSLOW: Will you discuss the amendment? The effect of the amendment is to provide that the four members named by the Council shall be non-chapter members.

MR. AVERY: I would like to raise a word of caution on that, because you will recall that my pet peeve has been that if we ever should have such a thing as members who do not belong to chapters, if we reward those members who do not belong to chapters with any such thing as this, I believe we kind of block ourselves from developing into a stronger chapter organization. Is it not sufficient again to leave these men who have devoted three or four years to Council service, to their judgment in picking these four men? If there are some worthy ones who should be considered and who do not belong to chapters, they could be selected without writing it into the By-Laws.

I tell you, gentlemen, you are trying to make this Constitution into an executive instrument and no Constitution should be so set up. You should leave something to your management, or else change the management.

PRESIDENT WINSLOW: Is there any further discussion of the amendment? If not, are you ready to vote on the amendment? Those in favor of the amendment will say *Aye;* Contrary-Minded? The motion is lost.

W. M. WALLACE, II, Durham, N. C.: I am not speaking for my chapter, but am voicing my personal opinion. I was talking yesterday to an executive of a large company and he pointed out to me that in a recent meeting they had changed their board of directors from six to 60. Someone asked him why in the world they would do such a thing as that and he said:

We have always thought that 60 people knew more than six and we have found in the last year operating under the 60 directors, that this group has not been unwieldy but quite satisfactory. We have received some very good suggestions from members who were not on the board of six.

For this reason I personally am against an amendment in any way. I think that with 37 men on the Nominating Committee, each of the chapters is represented and it is a pretty democratic way to do it. Any change from that method is only going to lead into the 37 members being required to be here to nominate seven, so that the seven will have seven votes for the 37; whereas, the Council has four members! It was something of this nature, I believe, that was amended in 1927. It was thought that it was not satisfactory at that time and now why should we want to go back to it. It is becoming a little bit alarming to me and to several other members with whom I have talked, that we have to go through this same rigmarole every year. It has been voted down at least twice and if it were possible I would like to make a motion, but I understand it is not, that it could not be brought up again for the next 10 years.

EDWIN ELLIOT, Philadelphia, Pa.: Speaking as an individual, I have listened to this discussion for I do not know how many years now. It has been brought up every year. Also, I have been an alternate at least, to the Nominating Committee on several occasions and I have seen how this thing works. The reason it is brought up every year is that the present setup is cumbersome and unwieldy. The point that has been lost sight of, I think, is the fact that these 37 men who come to the January meeting (as the delegates appointed by the Chapters to the Nominating Committee) invariably do not come to the Semi-Annual Meeting. If you had the same men come to the Semi-Annual Meeting then I would be in favor of continuing the present setup but as long as you do not, you do not have a coherent committee and that is why the present setup fails. By having a smaller Nominating Committee

and paying their traveling expenses, you are sure of having those same men at the summer meeting when the final nominations are made. That point I think should be borne in mind.

Another point that we should remember is the fact that this is not a political organization where there are a lot of plums to be shaken from the tree and there is no attempt or desire by anybody to grab control, or anything of that sort. We simply want to get a representative opinion of those men who are best fitted to hold office in the Society. At this meeting in January every chapter sends a delegate so that every chapter has a voice. After that chapter meeting they would then select a smaller and workable committee for nomination. I therefore am very heartily in favor of the amendment as it stands. It may have some inequalities. The geographical representation, this zoning, I do not think is important at all because it is not mandatory that a man be appointed from each of the seven zones. If you read it carefully you will find that no zone or district, however large it may be, can elect more than two, so you cannot have any large zone running away with the small ones. I think the proposition as a whole is very equitable.

E. K. CAMPBELL, Kansas City, Mo.: The Kansas City Chapter, after considering this amendment carefully, both through a special committee and at one of the chapter meetings, instructed me to vote No. It is also my personal preference that I should vote No. The only argument that bears any weight that I know of, is the confusion of such a large committee reaching a conclusion.

The Nominating Committee has been getting the job done ever since it was organized in 1927. The men have done a good job. I think there should be no change unless there is a positive reason for making such a change.

If it is in order after this vote is taken and this amendment loses, and you notice I say *after* and not *if*, I am going to offer a motion to the effect that this meeting should recommend to the Council that they pay the railroad fare of the present members of the Nominating Committee to the Semi-Annual Meeting. That will settle your whole problem and it will not come up for 10 years.

MR. LOCKE: I would like to add one thing, if I may. With all due respect to our fine Council, which we have had for a number of years and which has done outstanding work, it seems to me that we should not consider that at all, or tend toward personalities. That is not the intent of this group, I am sure. It is, I believe, a question of democratic procedure.

Of the 11 members of the Nominating Committee under this proposal, four will be chosen at one meeting and presumably those four will be very much of the same general type, shall we say. We then have seven other members chosen from all over the country from presumably seven zones. It would seem to me we would then have a committee of which four members would be most apt to think alike, and if the other seven tended to have any differences of opinion, the four would normally control, and that is my fundamental reason for opposition.

Requesting that only Members and Associate Members vote, President Winslow called for a rising vote and asked that H. J. Ryan and A. V. Hutchinson act as tellers. Following the vote President Winslow announced that the proposed amendment, which follows, had been passed by a vote of 81 to 47.

Proposed Amendment to Article B-VIII, *Section 11*[1]

Article B-VIII

Section 11, The Nominating Committee shall consist of eleven (11) members eligible to vote who have been in good standing continuously for five (5) years prior

[1] The proposed amendment is to be substituted for Article B-VIII, *Section 11* of the By-Laws.

to their selection. Four (4) members and one (1) alternate shall be chosen by majority vote of the Council at its last quarterly meeting of the calendar year. Seven (7) members and one (1) alternate shall be chosen by majority vote of the Chapter Delegates Committee on the first day of each Annual Meeting. The Secretary of the Society shall notify the Chairman of the Chapter Delegates Committee not later than December 1st of the names of the four (4) members and alternate selected by the Council.

Each regional area as defined in *Section 11A* must be represented on this Committee but not more than (1) member may be elected from any Chapter and not more than two (2) members may be elected from any regional area. Members of the Council are not eligible to serve on the Nominating Committee.

The Nominating Committee shall select eligible nominees for the office of President, First Vice-President, Second Vice-President, Treasurer, four (4) members of the Council and any other elective office that may have become vacant since the last annual election.

All members of the Nominating Committee who are present at the Annual Meeting shall be subject to the call of the Secretary of the Society on the last day of the Annual Meeting for the purpose of effecting its own organization and election of a permanent Chairman and Secretary. If a quorum is not available a temporary Chairman and Secretary shall be selected from the eligible Committee members and a permanent organization shall be established by correspondence or otherwise under the direction of the temporary Chairman. The Committee shall hold a formal meeting during the Semi-Annual Meeting of the Society for the final selection of nominees to which round trip railroad fare and lower berth expenses shall be paid by the Society to each Committee member and alternate attending. In case a Semi-Annual Meeting is not held the Committee must be called in a special meeting by the Chairman not later than September 1st of each year. In this event notice of such meeting shall be sent to all members and alternates by the Committee Secretary thirty (30) days prior to the date of such meeting. Expenses as previously outlined shall be paid to such special meeting.

Seven (7) members of the Nominating Committee shall constitute a quorum. Each member shall be entitled to vote for one (1) nominee for each office to be filled with the candidates receiving the highest number of votes receiving the nominations. The alternates may attend meetings of the Committee but shall have no vote except in the absence of one of the members with whom they are chosen.

The Nominating Committee shall certify the name of the nominees with their written acceptance to the Secretary of the Society prior to September 15th and the names of the candidates shall be published in the October JOURNAL of the Society.

Section 11A. Regional areas are defined as follows:

1. *The (New England) States of*
   Maine — New Hampshire
   Vermont — Rhode Island
   Massachusetts — Connecticut
   New York

2. *The (Middle Atlantic) States of*
   New Jersey — Delaware
   Pennsylvania — Maryland
   Ohio — Dist. of Columbia

3. *The (Middle Western) States of*
   Illinois — Michigan
   Indiana — Wisconsin

4. *The (Western) States of*
   Minnesota — North Dakota
   Iowa — South Dakota
   Nebraska — Montana
   Missouri — Wyoming
   Kansas — Colorado
   New Mexico

5. *The (Pacific Coast) States of*

Idaho	Washington
Utah	Oregon
Nevada	Arizona
	California

6. *The (Southern) States of*

Virginia	Mississippi
West Virginia	Kentucky
North Carolina	Tennessee
South Carolina	Louisiana
Florida	Arkansas
Georgia	Oklahoma
Alabama	Texas

7. *The Provinces of Canada*.

President Winslow then proposed consideration of the amendments to the Constitution and By-Laws which had been submitted to the membership on December 15, 1945, and requested that R. H. Carpenter, chairman of the Constitution and By-Laws Committee, make a statement on them.

Mr. Carpenter stated that the amendments to **Article C-XI,** *Section 1,* and **Article B-XI,** *Section 3,* were proposed in order to reduce the number of different funds within the Society. The creation of a building fund had led to the proposed change and the Finance Committee had recommended it as a means of facilitating the administration of Society funds.

Mr. Carpenter stated that the proposed amendment to **Article B-VIII,** *Sections 5, 8,* and *10,* served to define the functions of the Meetings and Publication Committees and to establish a Constitution and By-Laws Committee which would be able to act independently of the Council in proposing amendments. At the present time the Constitution and By-Laws Committee could consider only such amendments as might be originated by the Council and assigned to the committee.

Article B-XII, *Section 1,* was offered as a clarification of the present provision. It was moved by Mr. Carpenter, seconded by Mr. Blankin:

THAT the proposed amendments to the Constitution and By-Laws be adopted as follows:

Amendments to Constitution and By-Laws *

OF THE

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Article C-XI—Funds

Section 1—The collection, deposit, disbursement and investment of all funds of the Society shall be subject to the direction of the Council.

Article B-VIII—Committees

Section 5—The Meetings Committee shall select from the papers which have been approved by the Publication Committee or Committee on Research the papers for presentation at the technical sessions of the Annual and Semi-Annual Meetings, and shall assist the Publication Committee in securing desirable papers. Additional duties shall be as provided in the Rules.

* Prepared by the Committee on Constitution and By-Laws and approved by the Council.

Section 8—There shall be a Publication Committee of three (3) members appointed by the Council for a term of three (3) years, the term of one (1) member expiring each year. This Committee shall receive and examine all papers and written discussions intended for presentation by the Society, except those under the jurisdiction of the Committee on Research, and shall accept for publication such as it may approve. It shall approve for publication the Transactions of the Society containing the papers, discussions and abstracts of the Society and of the Council.

Section 10—At the first meeting of the Council after the Annual Meeting, the President shall appoint a Committee on Constitution and By-Laws which shall have supervision of all matters affecting the Constitution, By-Laws and Rules. This Committee shall consist of three (3) members, and shall report in writing to the Council on all actions taken which affect the Constitution, By-Laws and Rules.

Article B-XI—Funds

Section 3—An Endowment Fund for research and such other purposes devoted to the art of heating and ventilating and air conditioning, as determined by the Council, may be established. The interest or income from this fund shall be used each year in a manner and for purposes as determined by the Council. The principal shall remain intact and shall be deposited in banks or invested in a manner provided by *Section 1* unless otherwise directed by a vote of the membership conducted in the same manner as provided by **Article B-XVI.**

Omit *Section 7* **(Article B-XI—Funds)** and re-number *Section 8* to become *Section 7* and amend to read as follows: *Section 7*—A Society Reserve Fund shall be created into which all admission fees and such other monies as the Council may direct shall be placed until the fund totals a sum equal to fifteen dollars ($15.00) per member, or fifty thousand dollars ($50,000), whichever is smaller. This Reserve Fund is to be used only in cases of emergency when current Society revenues are not sufficient to pay necessary expenses. Withdrawals not to exceed twenty per cent (20%) of the fund in any calendar year may be authorized by the Society at any meeting provided it has been recommended by the Council. Investments shall be made in accordance with *Section 1* of **Article B-XI.** Interest earned on the Reserve Fund shall be added to current Society income.

Section 8—A Capital Fund shall be created in which all other monies shall be placed from which the funds for the Society and the Research operations shall be annually appropriated in accordance with the budgets approved by the Council.

Article B-XII—Publications and Papers

Section 1—A JOURNAL carrying papers approved by the Publication or Committee on Research, news items of Society interest, and such other material as may be decided by the Council, shall be published monthly.

President Winslow called for a vote and announced that the motion was carried.

Report of Tellers

Mr. Ryan, chairman of the Board of Tellers of Election offered the following report:

BALLOT OF OFFICERS

	For
President—Alfred J. Offner	1184
1st Vice President—Baldwin M. Woods	1186
2nd Vice President—G. L. Tuve	1186
Treasurer—J. F. Collins, Jr.	1186
Members of Council: E. G. Carrier	1186
F. W. Hutchinson	1185
R. A. Sherman	1185
M. S. Wunderlich	1183

Scattering votes for other candidates.

BALLOTS FOR COMMITTEE ON RESEARCH

L. N. Hunter	1186
C. O. Mackey	1186
R. D. Madison	1186
L. G. Miller	1186
L. P. Saunders	1186

BOARD OF TELLERS,
H. J. RYAN, *Chairman*
R. W. CUMMING
F. D. MCCANN

The meeting was adjourned at 4:00 p.m.

SECOND SESSION—TUESDAY, JANUARY 29, 9:30 A.M.

The second session convened at 9:45 a.m. with Pres. C.-E. A. Winslow, New Haven, Conn., presiding. R. C. Cross, Chicago, presented his paper, A Study of Several Coal-Fired Magazine-Feed Space Heaters, with H. N. Ostborg as co-author (see Chapter No. 1283).

President Winslow thanked Mr. Cross for a very helpful paper and then called for a presentation of the paper, The Application of High Temperature Liquid as a Heat or Energy Transfer Medium for Domestic and Industrial Uses, by O. O. Oaks, Raritan, N. J. (see Chapter No. 1284).

L. P. Saunders, Lockport, N. Y., treasurer, assumed the chair and presented Prof. F. W. Hutchinson, Lafayette, Ind., who gave an abstract of his paper, A Single-Equation Design Procedure for Radiant Panel Systems (see Chapter No. 1285).

The session was adjourned at 11:45 a.m.

THIRD SESSION—TUESDAY, JANUARY 29, 2 P.M.

The third session was called to order at 2:15 p.m. by President Winslow, who requested President-elect, Alfred J. Offner, New York, N. Y., to take the chair.

Dean John A. Goff, Philadelphia, Pa., presented an abstract of the paper, Low-Pressure Properties of Water from −160 to 212 F, with S. Gratch as co-author (see Chapter No. 1286).

The next paper, Therapeutic Uses of Low Temperature, was presented in abstract by the author, Frederick M. Allen, M.D., New York, N. Y. (see Chapter No. 1287).

B. M. Brod, LaGuardia Field, N. Y., presented an abstract of his paper, Heating and Ventilating for Transport Airplanes (see Chapter No. 1288).

The meeting was adjourned at 3:10 p.m.

FOURTH SESSION—WEDNESDAY, JANUARY 30, 9:30 A.M.

The fourth and final session convened at 9:45 a.m. with President Winslow presiding. President Winslow announced that the registration was 514 members, 134 guests, 77 ladies, a total of 725.

Prof. G. L. Tuve, Cleveland, Ohio, chairman of the Committee on Research, presented the report of the Committee on Research as follows:

Annual Report Committee on Research—1945

The resumption of peace-time activities has been gradual in Society Research as in other fields. But the research program has now more than attained its prewar size. Much actual research was initiated during 1945, and even more significant were several improvements in organization and preparation for an enlarged future program. The Director of Research has built up an excellent staff, and through active planning with the Technical Advisory Committees each research project is being closely related to the needs of the Society membership. A desirable correlation has recently been established with the Guide Publication Committee, so that research results may be more rapidly converted into useful Guide data.

An important event in the year's program was the assignment by the U. S. Public Health Service of a senior engineer to the Society's Research Laboratory at Cleveland, to conduct studies in the relationship of air conditioning and ventilation to industrial hygiene.

COMMITTEE ACTIVITIES

The Society is distinguished for its extensive research committee organization. Almost 300 committee memberships are involved in the research program. Omission of the Society's 1945 Semi-Annual Meeting cancelled the most important opportunity for mid-year committee meetings. But a large volume of work has been done by mail, and no less than a dozen special meetings of committees have been held. A like number are scheduled for the 1946 Annual Meeting.

The attention of Society members will be focused at an early date upon the activities of two special committees authorized by Council, the Committee on Research Financing, of which Dr. A. C. Willard is chairman, and the Committee on Housing, under the chairmanship of E. N. McDonnell, Chicago, Ill. An organization originally sponsored by the Committee on Research, but now independent of our Society, is the International Joint Committee on Psychrometric Data, of which J. Herbert Walker, Detroit, Mich., is chairman. This group is making an outstanding study of research data, which will be of great value to the profession. Participating in this joint effort are twelve scientific and engineering organizations of the United States, Great Britain and Canada.

REPORTS AND PLANS

A six-page *Research Reporter* was mailed in November to about 800 individuals and firms who have participated in financial or committee support of our research program. Plans for the 1946 program were briefly indicated, covering the nine projects in progress at the Laboratory in Cleveland, and 15 additional projects of ten cooperating institutions. About ten research papers have been completed during the year or are in preparation.

A budget of $110,000 per year for the next three years has been set as the goal for an adequate research program.

FUTURE NEEDS

The A.S.H.V.E. research program calls for three steps: *first,* defining the problems on which research is needed; *second,* getting the work done, and *third,* making the results available in useful form. The nature of our Society demands that research be highly diversified, and the Committee on Research recognizes this need. But to complete these three steps for each of a wide variety of research projects calls for the utmost in cooperation for each and every member of the Society. Following are outlines of certain urgent needs in carrying out each of the three steps:

1. ***Technical Suggestions:*** Criticisms of the research program should be translated by Society members into positive technical suggestions for research on specific topics. Critics are welcomed on the Technical Advisory Committees, to help plan a well-rounded program.

2. *Cooperative Projects:* Assistance is needed in securing more cooperating research institutions. This will both diversify the program and make it more economical. Supervision and overhead are usually assumed by the cooperating institution, thus adding both personnel and resources to the Society program.

3. *Publications' Program:* This is the most urgent need and the program should be supported by every member of the Society. It is suggested that research publications be of three types: (*a*) Each month the JOURNAL SECTION of the Society should carry short, practical research papers, giving data in useful and readable form. (*b*) Research Bulletins should be issued on all major projects, giving full data and results. Some of these would be reprints from the TRANSACTIONS. (*c*) A *Research Reporter* should be issued frequently, giving current news of the research program.

The Report of the Director of Research, Cyril Tasker, which follows, includes summaries of the work of the Technical Advisory and other Committees.

Report of the Director of Research

INTRODUCTION

The year 1945 has seen an expansion in the research activities of the Society, for, with an increased staff, we were able to activate some of the programs planned during the war years.

We were also able to secure both instruments and equipment and our inventory of such items shows a marked increase over that of a year ago. The installation of workshop machinery also enabled us to build many pieces of equipment at the Laboratory.

Staff

Clark M. Humphreys returned to the Laboratory on September 1st and took over the post of Senior Engineer. Herbert B. Nottage joined the staff on July 1st; his experience in the field of heat transfer opens up possibilities for further work in this field. Dr. Allen D. Brandt, assigned to the laboratory by the U. S. Public Health Service, joined us in October. This assignment marked the commencement of active participation by the Society in the fields of industrial hygiene and ventilation.

The additions to our senior staff necessitated additional clerical help; we have kept in close touch with most of the Technical Advisory Committees and the cooperating institutions and our contacts with other workers have increased greatly. We still need additional junior staff and technical assistants and hope to add these early in 1946.

The assistance rendered by all members of the staff in accelerating the research program during 1945 is hereby gratefully acknowledged, and the help given by the Secretary of the Society and the staff at the New York office is also acknowledged.

GUIDE ACTIVITIES

The coordination of research activities with other technical activities of the Society came much nearer to realization during 1945. The President appointed the Director of Research as a member of the Guide Publication Committee and every senior member of the Laboratory staff assisted in the revision of Guide chapters during 1945. Some chapters were wholly rewritten by staff members; others were reviewed and data accumulated by the Research Laboratory made available for THE GUIDE. This cooperation and coordination is of the greatest importance to the future of the Society since THE GUIDE enables the results of Society and other research to be translated into practice.

INDUSTRY CONTACTS, LIAISON AND PUBLICITY

There was a marked increase in the number of visitors to the Laboratory. Some of the Technical Advisory Committees met there in October, and many members visited the Laboratory at other times and discussed the programs under way. We also had visitors from England, official and unofficial, from Australia, Russia, and Sweden, while officers from the Canadian Department of Reconstruction made two visits to discuss our work on solar radiation transmission.

Though contacts by correspondence have greatly increased, we still lack the means whereby our research work may be effectively publicized and the results made known, in easily readable form, to every member of the Society and to the wider audience throughout the industry and among the general public. An attempt along these lines was made in October when a four-page pamphlet, *The Research Reporter*, with a one page insert, *Research in Progress*, was prepared at the Laboratory and distributed to some 400 organizations in the industry, to all officers of the Society, to Committee members and officers of local chapters, and to the entire membership.

STUDIES AT THE LABORATORY AND COOPERATING INSTITUTIONS

Rather than discuss the progress made at the Laboratory and the cooperating institutions according to the particular Technical Advisory Committees concerned, we have grouped the studies under a number of general headings. It is impossible, in a report of reasonable length, to give much beyond the barest outline, but the Laboratory staff will be pleased to supply further details to members and others interested, if they will write or visit us.

AIR FLOW, DISTRIBUTION AND FRICTION

In view of the importance to the industry of the items that may be classed under this general heading, it is encouraging to be able to report substantial activity during 1945. A sub-committee of the Technical Advisory Committee on Air Distribution and Air Friction met in March and formulated a program to cover a study of air friction in ducts which resulted in a paper[2] published by the Society. The new chart was submitted for acceptance by the Society and was published in THE GUIDE 1946. It represents a considerable advance over charts previously used and is based on recent work in the field of fluid mechanics which produced data of high accuracy having general acceptance.

Other aspects of the overall problem of air friction in ducts and fittings such as comparisons for round, square and oblong ducts, friction through flow-in branches, take-off from trunks, etc., are receiving attention and additional results should be available during 1946.

Barometric Dampers: Studies of barometric dampers under the Technical Advisory Committee on Fuels were enlarged to include studies of the flow of gases in a duct-tee section containing a swinging damper. Studies of the torque-displacement for representative dampers have been completed and a progress report prepared; studies on the elements of the flow system are now in progress.

Effect of Branch Take-off Design on Noise in Ventilating Duct Systems: The results of a study made for the Navy Department during the war were presented at the 52nd Annual Meeting in a paper by Oscar Imalis, the late F. C. Houghten, and Clark M. Humphreys (see Chapter No. 1289). The tests showed that: (*a*) The noise created by the splitter in a branch take-off arises from the turbulence of a relatively stationary mass of air back of the splitter. The noise can be reduced by properly streamlining the splitter. (*b*) Universally accepted standards of sound measurement techniques are badly needed in many phases of the ventilating and air conditioning industry and until such standards are available, the proper interpretation of sound measurements, and the correlation of the results of various investigators in the field, will be difficult or impossible.

Cooperative Research

1. *Case Institute of Technology:* The performance of diffusing type air outlets including wide-angle grilles and several sizes and types of perforated panels has been studied. Results, expressed in terms of dimensionless ratios, gave consistent curves when the residual velocity of the primary stream, expressed as a percentage of outlet velocity, was plotted against the distance from the discharge face in

[2] A.S.H.V.E. RESEARCH REPORT No. 1280—A New Friction Chart for Round Ducts, by D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 303.)

equivalent diameters. Instrumentation continues to be the major problem, and a better means for measuring room-air velocities is urgently needed.

Further work is now being done on the slot-type air outlet. With lower ceilings and little space for duct work and mixing zones, the location of multiple outlets on a long duct is becoming more common. Uniformity of air flow from all outlets of such an installation is a problem, whether the outlets are grilles, ceiling units, perforated panels or simple slots. Hence the distribution of air to long slots or multiple outlets is also being studied.

2. *Kansas State College:* These studies were planned to determine the characteristics of downwardly projected streams of heated air, to study the pattern of flow and to set up practicable means for predicting the maximum downward travel of the air when the surrounding room air temperature, orifice temperature of the projected air, size and shape of orifice, and velocity of projected air at the orifice are known. The buoyancy force which resists the downward flow of heated air considerably complicates the problem.

Mathematical analyses have been made and experimental work carried on to obtain temperature and velocity traverses, at different levels, across a stream of heated air projected from an orifice located 23 ft above the floor. This work is actively in progress and we are indebted to Prof. Linn Helander for his keen interest in the problem.

INDUSTRIAL (PROCESS) VENTILATION

Since Dr. Brandt took up his assignment, substantial progress has been made in the task of collecting and correlating all available information on process ventilation from engineering schools, governmental industrial hygiene departments, insurance companies, equipment manufacturers, consulting engineers, etc. The replies to the questionnaires distributed serve to show how little detailed, factual and dependable information is available.

All existing codes, laws and ordinances relating to the control of occupational diseases are being collected with a view to their correlation and the preparation of a model code.

At the Laboratory a start has been made on the study of air flow into suction openings to determine the center line flow equation for small, plain, freely suspended hoods.

Air Sterilization and Odor Control: The Committee under Prof. W. F. Wells, formed in 1944, has been working with a sub-committee of the *American Public Health Association* to set up a procedure for making sanitary air surveys. With the background of recent studies made for the Armed Services, the Committee plans to proceed at once with studies in schools, auditoria, etc.

HEAT FLOW AND HEAT TRANSFER

Under this general heading are classed studies under the Technical Advisory Committees on Glass, Heating Load, Cooling Load, Heat Transfer Surfaces, and Insulation. The comprehensive report, Heat Transmission through Windows and Glass Panels, prepared by G. V. Parmelee in 1944 has been under careful review by an editorial sub-committee prior to its publication as a special research bulletin.

Film Coefficients for Flat Surfaces: Studies have been in progress during 1945 using equipment designed, and for the most part constructed, at the Laboratory. The main features are a draw-through type wind tunnel and a plate, four feet in length, which can be heated electrically by sections. Much of the experimental work has been completed and a report, Forced Convection Heat Transfer Coefficients along a Flat Surface, distributed for Committee study. A second report entitled, Relationship Between Skin Friction of a Smooth Flat Plate and Heat Transfer Coefficients, is nearing completion. Continuing studies are planned to determine the effect of widely spaced transverse ribs, (*e.g.,* muntin bars on a window).

Solar Radiation Transmission through Glass: A comprehensive program recommended by the Glass Committee was endorsed by the Technical Advisory Committee on Cooling Load. Special apparatus was designed, and construction com-

menced in the early summer. It consists essentially of a 4 ft x 4 ft window frame pivoted on a horizontal axis, the whole being carried on a turntable. Heat transmitted by the window (or other glass surface) is absorbed by a water cooled section fastened to the rear side of the frame and the mechanism makes it possible to study the effect of the angle of incidence as well as the effects of both exterior and interior shading. Instrumentation and calibration are now under way; the equipment will be mounted in a suitable outside location so as to make studies under actual weather conditions.

Parallel with the design and construction of this apparatus a mathematical analysis was made of solar heat flow through all types of single and double glass and the results [3] were published in JOURNAL SECTION.

New tables of solar radiation values, tables of the direct solar heat transmitted through single standard glass for various latitudes, and tables giving new data on shading factors were prepared for inclusion in the chapter dealing with Cooling Loads in THE GUIDE 1946.

Heat Transfer of Refrigerants: At Case School of Applied Science (now Case Institute of Technology) refrigerant-side heat transfer coefficients in duct coils and unit coolers are being studied by three methods. A large number of full-sized coils have been tested and individual air-side and refrigerant-side coefficients determined. Two other methods, however, have been found more effective for studying the heat transfer of the evaporating refrigerant. A long-tube counterflow evaporator was first constructed to study the evaporation in straight tubes. Then return bends were added. Finally, an electrically heated tube has been set up, using currents of 200 to 1200 amperes passed through the tube itself. Effects of return-bends are clearly shown by the temperature-gradient curves. Coefficients of heat transfer for evaporation of *Freon-12* with no special equipment for separating oil have been observed to range usually from 100 to 500 Btu per square foot per degree difference between saturation temperature and surface temperature, depending largely on load.

Periodic Heat Flow: The mathematical analyses of heat flow through homogeneous walls and roofs, under way at Cornell University, were considered sufficiently complete to allow their being extended to cover composite walls and roofs. A report on this phase is now before the Committee on Cooling Load. Prof. C. O. Mackey has also prepared a report on The Sol-Air Thermometer—a New Instrument (see Chapter No. 1298), and is working on a simple method of presenting the essence of these studies for the determination of summer cooling loads within reasonable limits of accuracy. Subcommittes have been set up under the Committee on Cooling Load to deal with Appliances and Lights and Infiltration, and arrangements have been made to correlate all published work on periodic heat flow through homogeneous and non-homogeneous walls. Much remains to be done before usable, adequate data on periodic heat flow can be made readily available for the industry.

Heating Load and Infiltration: At the University of Illinois a survey of data taken in the I = B = R Research Home was made in an attempt to evaluate the effect of wind on the actual heat losses from the house. Fuel consumption curves for periods of the same average wind velocity were plotted against a range of average outdoor temperatures. In general the results indicated that for given indoor and outdoor temperatures the heat loss increased with increasing wind velocity up to a maximum velocity of about 9 mph. Between 9 and 15 mph the wind velocity seemed to have very little effect on the heat loss of the building. However, it was not possible to evaluate what portion of the total change in heat loss was a result of a change in infiltration rate and what portion was a result of change in the actual heat transmission through walls and glass. Other factors further complicated the analysis. Almost all published work to date serves to indicate how little we know about infiltration and how difficult is the problem of determining its effects on the heating load.

Insulation: Under the Committee on Insulation, a program has been initiated to obtain more accurate and dependable values of the thermal conductivity of insulating materials. All laboratories having equipment which conforms to the latest *ASTM*

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[3] A.S.H.V.E. RESEARCH REPORT No. 1281—The Transmission of Solar Radiation Through Flat Glass Under Summer Conditions, by George V. Parmelee. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 317.)

standards are being asked to cooperate in determining the thermal conductivity of test specimens of insulating material with the purpose of approving those laboratories whose results check those obtained at the Bureau of Standards on the same sample. With a number of approved laboratories available, manufacturers may have their products tested under standardized conditions and users of the data thus obtained will have confidence in the results.

A final report on the studies at Oregon State College on Heat Losses Through Wetted Walls (see Chapter No. 1300) is in preparation. In general, the results indicated that with a wet wall and an air velocity of 15 mph the average coefficient U for the wall under test was 0.318, an increase of 12 per cent over the coefficient for the dry wall with a 15 mph wind, and an increase of almost 32 per cent over that for the dry wall under still air conditions.

ENVIRONMENT, COMFORT, HEALTH AND EFFICIENCY

Navy Studies: The results of two studies made for the Navy Department during the war were released for publication and are being presented at the 52nd Annual Meeting.

The first is An Experimental Investigation of the Effect of Change in Atmospheric Conditions and Noise upon Performance (see Chapter No. 1291) by Dr. Morris S. Viteles, Department of Psychology, University of Pennsylvania, and Dr. Kinsley R. Smith, Department of Psychology, Pennsylvania State College. The tests were designed to ascertain the effects of various atmospheric conditions and noise levels on the accuracy, variability and volume of work that young men could perform in confined spaces on naval ships. The findings suggested that there was no need to provide equipment for the reduction of Effective Temperatures below the 80 deg level, with some evidence that it may be undesirable to attempt work, of the type studied, at Effective Temperatures as high as 87 deg. It was also indicated that it was unnecessary to spend large sums to reduce noise to below the 90 db level. In general it was shown that the effect of high temperatures on performance is much more adverse than that of high levels of noise.

The second paper, Physiological Response of Subjects Exposed to High Effective Temperatures and Elevated Mean Radiant Temperatures (see Chapter No. 1290), by Clark M. Humphreys, Oscar Imalis and Carl Gutberlet gives data indicating that within a wide range of environmental conditions all variables in a man's environment which affect his physiological reactions may be expressed in terms of Effective Temperature and mean radiant temperature elevation. The studies also showed that the effect of radiant heat on physiological reactions decreases as the Effective Temperature increases and becomes surprisingly small as the upper limits of endurability are reached. By repeated exposure to hot environments a person can acquire a considerable degree of acclimatization.

Other Studies: There have recently been made available several reports of studies made during the war for and by the Armed Services of the United States and its allies. Some contain information of direct interest and importance to heating, ventilating, and air conditioning engineers and preliminary steps have been taken towards an analysis of these reports with a view to summarizing their practical implications. The task is by no means easy but the results should justify the time and expense involved. Enormous sums of public money have been spent and there is no better body than the Society to make the results available for practicing engineers.

RADIANT HEATING AND COOLING

Studies on radiant heating and cooling have, unfortunately, had to take a back place during the past two years because of the demand of high priority war studies.

The cooperative research work at the University of California has resulted in five reports published in the A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning* in 1942, 1943 and 1944. In these the relationships existing between mean radiant temperature, surface temperature, shape moduli and related factors were discussed. In a paper, A Single-Equation Design Procedure for Radiant Panel Systems (see Chapter No. 1285) to be presented at the 52nd Annual Meeting, Prof. F. W. Hutchinson reviews the complex rational design procedure presented in

previous papers and proposes a single-equation design procedure for radiant heating systems.

A cooperative research project was initiated at Kansas State College during the year, the long range purpose of which is to study the relation between radiant heating and cooling, and human comfort and well-being. The immediate limited objectives are: (*a*) To determine the thresholds of human comfort when the boundary surface temperatures of a room are one of the controllable, variable factors contributing to human comfort or discomfort, and (*b*) To investigate means for determining mean radiant temperatures.

A psychrometric room has been designed and partially constructed, and most of the equipment necessary for operating the room and obtaining test data secured. It is planned to study the effect of cold and warm spots in walls, floors and ceilings and the manner in which one may compensate for such non-uniformities in surface temperatures.

At the Research Laboratory we have concentrated on the design of our new psychrometric rooms and are ready to proceed with their construction as soon as circumstances permit. A number of those participating in the research program have earmarked their contributions for studies in radiant heating.

MISCELLANEOUS STUDIES

1. *Psychrometry:* At the University of Toronto, studies have been under way on the measurement of the dew-point of air-water vapor mixtures and the development of a dew-point hygrometer to provide a standard for the correlation of psychrometric data. This is a piece of fundamental research work but the results may have wide practical applications for it is generally agreed that our present methods of psychrometric instrumentation are far from satisfactory. An extensive review of the literature was made prior to the commencement of experimental work.

2. *Temperature Measurement:* A short paper entitled, The Shielding of Thermocouples from the Effects of Radiation (see Chapter No. 1292), was prepared for presentation at the 52nd Annual Meeting. The tests indicated that random air currents can greatly affect the stability of readings of fine wire thermocouples in the presence of sources of radiation and that even shielding may not be wholly satisfactory. An aspirated thermocouple gives the true air temperature under most conditions.

Flow of Fluids: Studies are in progress at Texas A. and M. College to determine the increase in the friction in a stream of water flowing in a pipe and resulting from additional or secondary turbulence produced in the stream of water at the entrance of the pipe; for example, when, in a central hot water heating system, a 6-in. circulating pump is connected to a 12-in. flow main. In the present studies 4-in. pipe is being used and the connections between the pump and the pipe are being varied in size from ¾ in. to 2 in.

In the preceding paragraphs we have mainly discussed those studies which have progressed far enough during 1945 for papers to be prepared for presentation before the Society or for progress reports to be available for Committee study. Much additional work has been done and many other Committees have carried their studies to the stage where Laboratory projects can be initiated early in the year. The Committee on Air Cleaning is studying the very difficult problem of standard tests for air cleaning devices and finding no general agreement as yet on the composition of a standard dust for this purpose. The Committee on Corrosion has undertaken to prepare a chapter on corrosion for THE GUIDE, 1947. The Committee on Sorbents has gathered a great deal of data collected on the subject and is now correlating it for GUIDE use. The Committee on Heat Transfer Surfaces has tentatively approved a comprehensive program of fundamental research work on heat transfer of finned tubes, but important details are still to be worked out. The Committee on Weather Design Data has agreed on the basis for a statistical analysis of a great deal of data collected a few years ago, and plans to make the analysis as soon as competent personnel and sorting equipment are available.

The work ahead of us is sufficiently well defined to ensure a continuous program during 1946—some new projects will be set up early in the year and many of the projects active during 1945 will be brought to the report stage during 1946.

STATEMENT OF RESEARCH INCOME AND EXPENSES

(For the Fiscal Year Ended October 31, 1945)

INCOME

FROM A.S.H.V.E.				
40% of 1945 and prior years dues collected			$24,630.65	
Special Council appropriation			10,000.00	$34,630.65
CONTRIBUTIONS—PER SCHEDULE				
General			$ 8,361.59	
Earmarked			—0—	
Contributions received	$11,450.00			
Add: Deferred from prior year	1,033.25	$12,483.25		
Less: Deferred to future operations		3,063.37	9,419.88	
Navy Contract			1,600.00	19,381.47
INTEREST FROM BANK DEPOSITS				50.82
TOTAL INCOME				$54,062,94

EXPENSES

CHAIRMAN AND COMMITTEE			
Traveling	$ 608.02		
Chairman's Office	340.45		
Professional Services	250.00	$ 1,198.47	
RESEARCH LABORATORY			
Salaries—Administrative and Technical	$21,490.52		
Salaries—Clerical	5,088.60		
Traveling	1,862.41		
Postage	362.51		
Telephone and Telegraph	621.30		
Office Expenses and Supplies	585.03		
Printing and Mimeographing	572.72		
Library and Periodicals	255.99		
Laboratory Materials and Supplies	3,465.83		
Depreciation—Equipment and Fixtures	1,239.37		
Unallocated	337.77		
Taxes (Cleveland rent)	2,180.00		
Property Insurance	168.40		
Heat, Electricity, Gas, Water	1,109.71		
Janitor	1,274.09		
Building Maintenance	83.70	40,697.95	
COOPERATIVE RESEARCH			
University of California	$ 800.00		
University of Minnesota	900.00		
Case Institute of Technology	2,000.00		
Oregon State College	400.00		
Texas A and M College	375.00		
Kansas State College	1,833.33		
University of Toronto	1,307.20		
University of Illinois	300.00		
State University of Iowa	333.33	8,248.86	
NAVY CONTRACT		1,600.00	
SPECIAL COUNCIL APPROPRIATION			
Expenditures for Laboratory Equipment, etc., charged against Council Appropriation		5,862.42	57,607.70
EXCESS OF EXPENSES OVER INCOME			$ 3,544.76

Participating in 1945 Research Program

Aerofin Corp.
** Airtemp Div. of Chrysler Corp.
* AGA Testing Laboratories
Allegheny County Steam Heating Co.
Aluminum Co. of America
** American Blower Corp.
* Anemostat Corporation
A.S.H.V.E.-Michigan Chapter
Barnes & Jones, Inc.
Bethlehem Steel Co.
* Blue Ridge Glass Corp.
** Buffalo Forge Co.
Burnham Boiler Corp.
Brunner Manufacturing Co.
** Byers Co., A. M., The
Carrier Corp.
** Chamberlin Co. of America
Chicago Pump Co.
Chase Brass & Copper Co.
Crane Corp.
Dallas Engineering Co.
* Detroit Edison Co.
Dole Valve Co.
Duquesne Light Co.
Dunham Company, C. A.
Elliott Electric Co.
Heating, Piping and Air Conditioning Contractors National Association
Forslund Pump & Machinery Co.
Frick Co., Inc.
* Friez Instrument Co. (Bendix Corp.)
G & O Manufacturing Co.
Graphite Products Corp.
Hays Corp.
Hoffman Specialty Co., Inc.
* Holcomb & Hoke Mfg. Co.
Ilg Electric Ventilating Co.
Illinois Engineering Co.
Iron Fireman Manufacturing Co.
Jenkins Bros.
Johnson Service Co.
** Johns-Manville Co.
Kinetic Chemicals, Inc.
Keeney Publishing Co.
Kewanee Boiler Corp.
* La-Del Conveyor & Mfg. Co.
* Libbey-Owens-Ford Glass Co.
Mahon Company, R. C., The
Marsh Company, The
McDonnell & Miller, Inc.
McQuay, Inc.
Mellish & Murray Co.
* Narowetz Heating & Ventilating Co.
National Radiator Co.
National Mineral Wool Association
Nash Engineering Co., The
** Nesbitt, John J., Inc.
* Owens Corning Fiberglas Corp.
* Perfex Corp.
* Pittsburgh Plate Glass Co.
Pipe Fabrication Institute
Portland Cement Association
Pyle-National Co., The
Raisler Co., The
* Russell, F. C., Co., The
* Sarco Co., Inc.
* Sarcotherm Controls, Inc.
Sisalkraft Co., The
Spencer Thermostat Co.
Sturtevant, B. F., Co.
Star Electric Motor Co.
Surface Combustion Corp.
Taylor Instrument Companies
Timken-Detroit Axle Co.
* Trade-Winds Motorfans, Inc.
U. S. Steel Corp.
Warren Webster & Co.
Weil-McLain Co.
Webster Engineering Co.
* Webster Electric Co.
Wheeler, W. H., Inc.
Williams Oil-O-Matic Heating Corp.
Wing, L. J., Mfg. Co.
Wood Conversion Co.
* Worthington Pump & Machinery Corp.
York Corp.
* Young Radiator Co.

* Earmarked Participation.
** Combination of Earmarked and General Participation.

RESEARCH PAPERS—1945

1. The Transmission of Solar Radiation Through Flat Glass Under Summer Conditions, by George V. Parmelee (Cleveland) (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 317).
2. A New Friction Chart for Round Ducts, by D. K. Wright, Jr. (Cleveland) (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 303).
3. An Experimental Investigation of the Effect of Change in Atmospheric Conditions and Noise Upon Performance, by Morris S. Viteles, Ph.D. (Philadelphia) and Kinsley R. Smith, Ph.D. (State College) (see Chapter No. 1291).
4. Physiological Response of Subjects Exposed to High Effective Temperatures and Elevated Mean Radiant Temperatures, by Clark M. Humphreys (Cleveland), Oscar Imalis and Carl Gutberlet (Pittsburgh) (see Chapter No. 1290).
5. Effect of Branch Take-Off Design on Noise in Ventilating Duct Systems, by Oscar Imalis (Pittsburgh), F. C. Houghten (deceased) and Clark M. Humphreys (Cleveland) (see Chapter No. 1289).
6. The Shielding of Thermocouples from the Effects of Radiation, by George V. Parmelee and Richard G. Huebscher (Cleveland) (see Chapter No. 1292).

INSTITUTIONS COOPERATING WITH THE COMMITTEE ON RESEARCH

Agricultural and Mechanical College of Texas, College Station, Texas: The Effect of Secondary Turbulence on the Friction in a Flowing Stream of Water. *Case Institute of Technology,* Cleveland, Ohio: Air Distribution in Rooms; Heat Transfer Coefficients of Freon Refrigerants. *Cornell University,* Ithaca, New York: Periodic Heat Flow Through Composite Walls or Roofs. *Oregon State College,* Portland, Oregon: Heat Losses Through Wetted Walls. *Kansas State College,* Manhattan, Kansas: Projection of Heated and Cooled Air Streams; Experimental Studies of Radiant Heating and Cooling. *University of California,* Berkeley, California: Air and Surface Temperatures in a Panel-Heated Room; The Measurement of the Radiant Effect in Heated and Cooled Rooms; Cooling Tower Design and Performance. *University of Illinois,* Urbana, Illinois:

Heat Losses Due to Infiltration. *University of Minnesota*, Minneapolis, Minnesota: Statistical Analysis of Variables Entering into Sensory Reactions to Summer Air Conditioning. *University of Toronto*, Toronto, Ontario, Canada: Moisture Condensation on a Surface and the Measurement of the Dew-Point.

President Winslow paid tribute to Professor Tuve, Chairman of the Committee on Research, for the results which he had accomplished in financing and establishing an adequate Laboratory, and in developing a comprehensive research program during a trying period.

C. M. Humphreys presented his summary of the paper, Effect of Branch Take-Off Design on Noise in Ventilating Duct Systems, by Oscar Imalis, F. C. Houghten, and C. M. Humphreys, which was prepared by Dr. Houghten and Dr. Imalis at the Pittsburgh Laboratory (see Chapter No. 1289).

An abstract of the next paper, An Experimental Investigation of the Effect of Change In Atmospheric Conditions and Noise Upon Performance, by Morris S. Viteles and Kinsley R. Smith, was presented by Dr. Viteles (see Chapter No. 1291).

Dr. Winslow remarked that this paper was an extremely valuable and interesting contribution not only to the practical problem in which the Navy was interested but also to general physiological knowledge.

Mr. Humphreys presented an abstract of the third paper, Physiological Response of Subjects Exposed to High Effective Temperatures and Elevated Mean Radiant Temperatures, by C. M. Humphreys, Oscar Imalis, and Carl Gutberlet (see Chapter No. 1290).

The fourth and final technical paper, Shielding of Thermocouples from the Effects of Radiation, by G. V. Parmelee and R. G. Huebscher (see Chapter No. 1292), was presented in abstract by Mr. Parmelee.

Dr. Winslow stated that the Council had adopted appropriate resolutions of condolence regarding the loss of members during the past year by death and referred to Mrs. Olive E. Frank, one of the Society's first women members who died on January 28.

President Winslow also announced that Prof. Clyde A. McKeeman, formerly of Case Institute of Technology would start on February 1 as assistant to the president of the Society and would direct the fund raising for research.

The president called for unfinished business and, as there was none, the meeting was declared open for new business.

Dr. B. M. Woods, Berkeley, Calif., referred to the Amendments to the Constitution and By-Laws which had been passed at the first session on January 28. He stated that the new *Section 3* of **Article B-XI** specified procedure under which it was possible for the Society through its Council to invest funds from the endowment account in other than specified groups of securities. In order to purchase the laboratory, it would be necessary to withdraw funds from the investment account and permission to do so must be obtained by a vote of the membership after following the required 30 days' notice. Such a vote could probably not be taken before the Semi-Annual meeting in June. Dr. Woods had prepared the necessary resolution for presentation to the Society and requested that all of the members who were interested should sign the resolution, although the signatures of only 10 were actually required.

President Winslow stated that his attention had been directed to an important research project in the Society's field of interest which was under

way at one of the universities. He suggested that any member having such a project in mind confer with Professor McKeeman in order that the Society might take its proper part in such a research program.

C. E. Price, Chicago, presented the report of the Committee on Resolutions and moved the adoption of the following resolutions:

Resolutions

WHEREAS, the splendid success of the 52nd Annual Meeting of the A.S.H.V.E. has been due to the careful and thorough preparations of New York Chapter through its Committee on Arrangements under the able leadership of its chairman, R. H. Carpenter, and of H. J. Ryan, president of the chapter, and

WHEREAS, many organizations and individuals have participated in activities providing members and guests with opportunities for entertainment, inspection and instruction.

* * *

BE IT RESOLVED THAT an expression of appreciation be adopted and that copies be sent to each of the following:

To R. H. Carpenter chairman of the Committee on Arrangements and to his several committee chairmen and their ladies,
To H. J. Ryan, President of New York Chapter, A.S.H.V.E.,
To the authors for the interesting papers presented at the technical sessions,
To W. H. Driscoll, the banquet toastmaster,
To Harvey Wiley Corbett, the speaker at the Get-together Luncheon,
To The National Broadcasting Co. for tickets to radio broadcasts,

* * *

To the Columbia Broadcasting Co. for tickets to radio Broadcasts,
To The Roxy Theater for complimentary theater tickets.
To Sheffield Farms Co., Inc., for the opportunity to inspect milk processing in a modern dairy,
To the Metropolitan Life Insurance Co. for the opportunity to inspect the mechanical equipment of a large air conditioned building,
To the management and staff of the Commodore Hotel for their courtesy and service,
To the newspapers and trade papers for their coverage of the 52nd Annual Meeting.

The motion was seconded by W. T. Jones, Boston, and was passed unanimously.

INSTALLATION OF OFFICERS

President Winslow announced that the next item of business was the installation of officers and requested that W. T. Jones, past president conduct the ceremonies. Mr. Jones stated that Dr. Winslow while occupying the office of president had won the esteem and respect of the entire membership by his keen mind, his fairness as a presiding officer, and his outstanding ability.

A. J. Offner was installed in the office of president. Other incoming officers installed were: 1st Vice President B. M. Woods by Past President S. H. Downs; 2nd Vice President G. L. Tuve by Past President M. F. Blankin; Treasurer J. F. Collins, Jr., by Past President W. H. Driscoll.

The new members of the Council installed by Past President W. L. Fleisher were: E. G. Carrier, Boston; F. W. Hutchinson, Lafayette, Ind.; R. A. Sherman, Columbus, Ohio; and M. S. Wunderlich, St. Paul, Minn.

Mr. Jones then declared the new group of officers properly installed and turned the gavel over to President Offner.

As his first official act President Offner declared the 52nd Annual Meeting adjourned at 12:00 noon.

WELCOME LUNCHEON

At the opening luncheon, Monday afternoon, R. H. Carpenter acted as toastmaster. The speaker, Harvey Wiley Corbett, prominent New York architect, addressed 330 members and guests, on the subject, Are We Over Mechanizing Life. Mr. Corbett discussed the problems of constructing build-

ings, their equipment and the operation and control of the mechanical devices that are installed.

BANQUET

The annual banquet was held Tuesday evening, January 29, with W. H. Driscoll serving as toastmaster.

During the banquet, Capt. A. E. Stacey, Jr., was presented the F. Paul Anderson Medal for his contributions to heating, ventilating and air conditioning; Dr. C.-E. A. Winslow, retiring president, received the Past President's Emblem; and S. H. Downs, president of the Society in 1944, was presented with the Past President's Memory Book. A special presentation was made to Mr. Driscoll to commemorate his having been a member since 1904, during which time he had served as president of the Society, and of the New York Chapter, and also to commemorate the 50th year of his entry into the field of heating, ventilating, and air conditioning.

The speaker of the evening was Leland Stowe, well known journalist, foreign correspondent and author, who spoke on What We May Expect in the Future, and particularly the problems of the atomic age.

PROGRAM 52ND ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL COMMODORE, NEW YORK, N. Y.
JANUARY 27-30, 1946

Sunday, January 27

1:30 P.M. REGISTRATION—(Parlors B and C—Ballroom Floor)
2:00 P.M. Council Meeting—(Parlor G)

Monday, January 28

9:00 A.M. REGISTRATION—(West Ballroom Foyer)
10:30 A.M. Inspection trip to Metropolitan Life Office Building Air Conditioning Plant
12:15 P.M. Welcome Luncheon—(East Ballroom)
Toastmaster: R. H. Carpenter
Speaker: Harvey Wiley Corbett
Subject: Are We Over Mechanizing Life?
2:00 P.M. BUSINESS SESSION—(West Ballroom)
Greetings: H. J. Ryan, President New York Chapter
Reports of Officers and Council Committees
Report on Revision of Charter
Report on Nominating Committee Amendment
Amendments to Constitution and By-Laws
Report of Tellers of Election
2:30 P.M. Ladies Card Party and Tea
Empire State Tower Visit
6:30 P.M. Mardi-Gras Party—Dinner and Carnival in the Hotel Capitol—51st St. at 8th Avenue. A lavish revue staged by John Murray Anderson

Tuesday, January 29

9:00 A.M. REGISTRATION—(West Ballroom Foyer)

9:30 A.M. TECHNICAL SESSION—(West Ballroom)
A Study of Several Coal-Fired Magazine-Feed Space Heaters, by R. C. Cross and H. N. Ostborg
The Application of High Temperature Liquid as a Heat or Energy Transfer Medium for Domestic and Industrial Uses, by O. O. Oaks
A Single-Equation Design Procedure for Radiant Panel Systems, by F. W. Hutchinson

10:00 A.M. Inspection of Show Plant of Sheffield Farms
Trip to Statute of Liberty

2:00 P.M. TECHNICAL SESSION—(West Ballroom)
Low Pressure Properties of Water from —160 to 212 F, by John A. Goff and S. Gratch
Therapeutic Uses of Low Temperature, by Frederick M. Allen, M.D.
Heating and Ventilating for Transport Airplanes, by B. M. Brod

2:00 P.M. Sightseeing and Museum Visit

7:00 P.M. Annual Banquet—(Grand Ballroom)
F. Paul Anderson Medal presented to Capt. A. E. Stacey, Jr., by Dr. W. H. Carrier
Past President's Emblem presented to Dr. C.-E. A. Winslow, by W. L. Fleisher
Past President's Memory Book presented to S. H. Downs, by Dr. C.-E. A. Winslow
Speaker: Leland Stowe, Journalist, Pulitzer Prize Winner, Foreign Correspondent and Author
Address: What We May Expect in the Future

Wednesday, January 30

9:30 A.M. TECHNICAL SESSION—(West Ballroom)
Report of Committee on Research
Effect of Branch Take-Off Design on Noise in Ventilating Duct Systems, by Oscar Imalis, F. C. Houghten, and C. M. Humphreys
Physiological Response of Subjects Exposed to High Effective Temperatures and Elevated Mean Radiant Temperatures, by C. M. Humphreys, Oscar Imalis, and Carl Gutberlet
An Experimental Investigation of the Effect of Change in Atmospheric Conditions and Noise Upon Performance, by Morris S. Viteles, Ph.D., and Kinsley R. Smith, Ph.D.
The Shielding of Thermocouples from the Effects of Radiation, by G. V. Parmelee and R. G. Huebscher
Unfinished Business
New Business
Installation of Officers
Adjournment

12:30 P.M. Council Luncheon Meeting—(Parlor A)

1:30 P.M. Inspection Trip to Metropolitan Life Building Air Conditioning Plant

SPECIAL ENTERTAINMENT

Sunday, January 27

Free tickets were distributed at Registration Desk for the following broadcasts: R.C.A. Show (Deems Taylor)—4:30 p.m.; Prudential Family Hour, and the Philharmonic Symphony—5:00 p.m.; Philip Morris Program, and Fred Allen Show—8:30 p.m.; Texaco Hour (James Melton)—9:30 p.m.; G. E. Hour of Charm (Phil Spitalny All-Girl Orchestra)—10:00 p.m.; We the People—10:30 p.m.; Philip Morris (Re-Broadcast)—11:00 p.m.

Monday, January 28

The Entertainment Committee arranged for free tickets for those who did not wish to remain at the Carnival for dancing, for the following performances:

ROXY THEATER (50th St. at 7th Ave.)

Stage Presentation 9:30 p.m.

Motion picture "Leave Her to Heaven" at 10:20 p.m.

Bob Hawk Broadcast (formerly "Thanks to the Yanks") at 10:30 p.m.

COMMITTEE ON ARRANGEMENTS

R. H. CARPENTER, *General Chairman*

Vice-Chairmen

M. C. GIANNINI — W. E. HEIBEL — H. S. JOHNSON — C. S. KOEHLER — H. J. RYAN

Banquet—ALFRED ENGLE, *Chairman;* W. A. SWAIN, *Vice-Chairman*; H. H. BOND, W. A. SHERBROOKE, J. H. WHITE

Entertainment—M. C. GIANNINI, *Chairman;* A. A. BEARMAN, A. F. HINRICHSEN, B. H. LEE, F. D. MCCANN, HOWARD J. ROSE, JOSEPH WHEELER, JR.

Finance—W. M. HEEBNER, *Chairman;* H. W. FIEDLER, *Vice-Chairman;* G. E. OLSEN

Inspection—C. S. PABST, *Chairman;* J. F. DAILEY, J. A. HELLER, F. D. LAWRENCE

Ladies—H. S. WHELLER, *Chairman;* MR. AND MRS. ROBERT L. BLANDING, R. W. CUMMING, MRS. ALFRED J. OFFNER, MR. AND MRS. T. R. PEYREK, MR. AND MRS. C. H. SMITH, W. G. TURNO, MRS. H. S. WHELLER

Publicity—C. F. ROTH, *Chairman;* CARL H. FLINK, W. J. OSBORN, R. V. SAWHILL, CLIFFORD STROCK

Sessions—E. J. RITCHIE, *Chairman;* THOMAS BAKER, H. S. BIRKETT, R. B. DARLING, C. R. HIERS, L. C. MEDCALF, K. L. MYTINGER, J. G. SKIDMORE

Reception—H. J. RYAN, *Chairman;* M. C. GIANNINI, W. E. HEIBEL, H. S. JOHNSON, C. S. KOEHLER

Secretary—CARL H. FLINK

1283

No. 1283

A STUDY OF SEVERAL COAL-FIRED MAGAZINE-FEED SPACE HEATERS

By R. C. Cross * and H. N. Ostborg,** Chicago, Ill.

WITHIN recent time, there has been a definite revival of interest in the development of coal-fired magazine-feed space heaters, and a number of designs of this type heater have been placed on the market. The general development program has been stimulated by the widespread use of these heaters in army establishments and, to a lesser extent, by their use for supplementary heating in homes in oil-rationed areas. A general demand by the public for the convenience advantage of the magazine-feed feature also has been a contributory influence.

The factors of prime importance in the performance of this type of heater are capacity, efficiency, and smoke-producing tendency. The data presented in this discussion are the results of an investigation to determine these factors as applied to a number of heaters of different makes.

Summary

The findings of this investigation can be summarized as follows:

1. There is considerable variation in the capacities and efficiencies of coal-fired magazine feed heaters of similar external dimensions. Five of the ten heaters tested failed to meet the 50 per cent efficiency, at rating, stipulated in Bureau of Standards TS-3443, Proposed Commercial Standard for Coal-burning Space Heaters.

2. High flue gas temperatures, and excessive losses in the dry flue gas indicate a general lack of sufficient flue travel, and secondary heating surface.

3. The majority of the heaters operated with low CO_2 and high excess air in the flue gases. These heaters were provided with various methods of introducing secondary air, and it would appear that these methods are not too successful.

4. The superior efficiency of the heaters utilizing the cross-feed, and the under-feed method of burning was noteworthy. In the case of the under-feed heater, however, the serious smoke-back nullified the general performance advantage.

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* Head, Mechanical and Combustion Div., Sears, Roebuck & Co., Member of A.S.H.V.E.
** Senior Testing Engineer, Sears, Roebuck & Co., Merchandise Testing and Development Laboratory.

Presented at the 52nd Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1946.

5. The average attention period encountered in the heaters tested was approximately 18 hours. This is a satisfactory value for achievement at rated output, TS-3443 stipulating a minimum attention period of 12 hours for magazine-feed heaters.

6. With those heaters provided with secondary air admission the smoke produced, when high-volatile fuel was used, was not excessive, except for the period immediately following the charging of fuel.

7. In general, the heaters reacted positively to changes in draft, and primary air damper setting.

8. The heater using the under-feed principle of burning showed a serious fault in strong smoke-back when the magazine charging door was opened. A fair amount of smoke-back was experienced with the cross-feed heaters.

DESCRIPTION OF HEATERS

The heaters tested were all of the magazine-feed radiant type, but they may be classified by the method of burning. Heaters A and J utilized a modified principle of cross-feed burning, with an inverted fuel bed. Heater F operated with under-feed burning with an inverted fuel bed, and the remainder of the heaters were of the conventional over-feed burning type. These latter heaters, although designated as *magazine-feed,* are truly surface-fired heaters incorporating deep firepots. The combustion chamber and so-called *magazine* are common, and magazine-feed cannot be applied in a generic sense.

For purpose of definition, it is noted that in cross-feed burning the direction of coal feed and flow of air for combustion are at right angles to each other. The coal feed and flow of air for combustion move in the same direction in under-feed burning. In conventional over-feed burning the coal feed and air flow move in opposite directions.

All of the heaters, except Heater F, provided some means for admitting secondary air into the combustion space. The effect of secondary air admission is noted in the discussion of the test results. Heaters C and H had metal liners in the combustion chamber, whereas all the other heaters were lined with refractory material.

TEST SET-UP

The general arrangement of the equipment and instruments used in obtaining the test data is shown in Fig. 1. In order to obtain the rate of burning directly by weight loss, the heater was set on a scale. The instruments for measuring drafts and temperatures, flue gas collection and analysis equipment, and a smoke recorder of the photoelectric type were connected to the heater in accordance with recognized test procedure.

TEST PROGRAM AND PROCEDURE

The test program was divided into two parts—a series of capacity, or rating, tests with anthracite, and a series of tests with high-volatile bituminous coal to determine the smoke producing tendencies and draft reaction characteristics of the heaters.

In the anthracite tests the recorded data included the weight of coal fired, weight of ash and refuse, ashpit draft, draft at the flue gas outlet, flue gas composition and temperature, temperature of the air used for combustion, and the rate of burning was determined by weight loss of the unit. In addition to these data, the proximate and ultimate analyses of the fuel and the combustible in the ash and refuse were determined. These data permitted the determination of the stack and grate losses in a conventional heat balance.

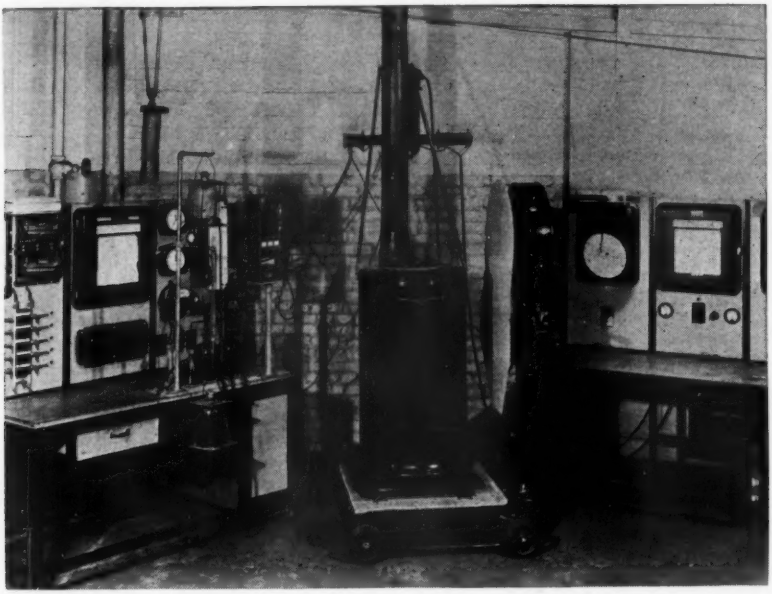

FIG. 1. GENERAL ARRANGEMENT OF EQUIPMENT

When burning anthracite there are no unaccounted for losses due to unburned hydrocarbons in the flue gases, and the heater output was obtained by subtracting the total stack and grate losses from the heat value of the coal.

A preliminary starting fire was used in all tests. During this preliminary period and the test period, the primary air damper was adjusted to provide a maximum rate of burning with the limitation of a maximum flue gas temperature of approximately 900 F. The draft at the flue gas outlet was maintained at 0.06 in. water, but because the rate of burning was regulated by the primary air damper adjustment, the output is not strictly the result of the maintenance of the constant draft value. The test period was considered ended when the flue gas temperature had dropped to 75 per cent of the average value obtained in the preliminary period.

TABLE 1—TEST DATA—ANTHRACITE SERIES

	A		B		C		D		E		F		G		H		J		K	
GENERAL INFORMATION																				
1. Heater designation																				
2. Type of burning	CROSS-FEED		OVER-FEED		OVER-FEED		OVER-FEED		OVER-FEED		UNDER-FEED		OVER-FEED		OVER-FEED		CROSS-FEED		OVER-FEED	
3. Length of test-hour	18.0		16.5		12.25		18.16		18.25		31.0		20.25		15.25		15.33		17.75	
WEIGHTS—POUNDS																				
4. Total weight loss	84.5		70.0		47.5		78.5		80.0		176.0		90.0		72.5		67.0		94.0	
5. Equivalent fuel burned	94.7		78.5		53.5		88.1		89.5		197.5		101.0		81.25		75.2		105.5	
ASH AND REFUSE																				
6. Weight of ash and refuse—lb	4.0		12.25		5.25		10.75		8.5		9.0		12.0		2.75		3.5		10.2	
7. Combustible in ash and refuse—per cent	0.0		31.0		4.5		10.5		0.0		0.0		9.2		0.0		0.0		0.0	
8. Carbon burned per lb fuel fired—lb	0.77		0.71		0.76		0.75		0.77		0.77		0.76		0.77		0.77		0.77	
DRAFTS—INCHES WATER																				
9. Ashpit	0.048		0.031		0.041		0.026		0.025		....		0.032		0.057		....		0.015	
10. Flue gas outlet	0.060		0.060		0.063		0.060		0.060		0.060		0.060		0.060		0.041		0.059	
FLUE GAS COMPOSITION, PER CENT																				
11. Carbon dioxide (CO_2)	6.7		6.8		5.2		4.6		5.3		11.6		8.0		6.9		10.7		5.7	
12. Oxygen (O_2)	13.7		13.6		15.1		16.3		15.2		6.3		12.5		13.4		9.6		14.7	
13. Carbon monoxide (CO)	0.0		0.0		0.0		0.0		0.0		3.0		0.0		0.0		0.0		0.0	
14. Nitrogen (N_2) (by difference)	79.6		79.6		79.7		79.1		79.5		79.1		79.5		79.7		79.7		79.6	
15. Dry flue gas per lb of fuel—lb	28.4		25.8		35.9		39.9		35.6		13.2		23.6		27.2		17.9		33.0	
16. Excess air—per cent	188.0		184.0		256.0		354.0		262.0		30.0		147.0		176.0		84.0		233.0	
TEMPERATURES—F																				
17. Air for combustion	72		82		86		81		73		86		75		80		96		85	
18. Flue gases	669		863		720		860		858		615		908		713		944		890	
HOURLY RATES																				
19. Fuel burned—lb/hr	5.26		4.76		4.36		4.85		4.81		6.37		4.93		5.32		4.9		5.94	
20. Heat release rate—Btu/hr	67,749		61,309		56,157		62,468		61,953		82,046		63,498		68,522		63,122		76,500	
HEAT BALANCE	Btu	%	Btu	%	Btu	%	Btu	%	Btu	%	Btu	%	Btu	%	Btu	%	Btu	%	Btu	%
21. Heat lost in steam in flue gases	470	3.6	505	3.9	496	3.9	518	4.0	518	4.0	474	3.7	527	4.0	492	3.8	526	4.1	563	4.4
22. Heat lost in dry flue gases	3342	26.0	4835	37.6	5470	42.4	7465	57.8	6715	52.0	1678	13.0	4725	36.7	4135	32.1	3655	28.4	6376	49.5
23. Heat lost in carbon monoxide	0	0.0	0	0.0	0	0.0	0	0.0	0	0.0	1602	12.5	0	0.0	0	0.0	0	0.0	0	0.0
24. Heat lost in combustible in ash	0	0.0	560	4.2	92	0.7	238	1.8	0	0.0	0	0.0	146	1.1	0	0.0	0	0.0	0	0.0
25. Total measurable losses	3812	29.6	5900	45.8	6058	47.0	8221	63.6	7233	56.0	3754	29.2	5398	41.8	4627	35.9	4181	32.5	6939	53.9
26. Calorific value of fuel	12880	100.0	12880	100.0	12880	100.0	12880	100.0	12880	100.0	12880	100.0	12880	100.0	12880	100.0	12880	100.0	12880	100.0
27. Heat utilized, efficiency	9068	70.4	6980	54.2	6822	53.0	4659	36.4	5647	44.0	9126	70.8	7482	58.2	8253	64.1	8699	67.5	5941	46.1
OUTPUT																				
28. Output—Btu/hr	47,688		33,200		29,743		22,596		27,259		58,088		36,956		43,923		42,600		35,250	

No manual attention, such as poking or shaking the grates, was given during the test period. In those heaters provided with independent control of secondary air, the secondary air inlet was set at the closed position.

The data obtained in the tests using high-volatile bituminous coal included the weight of coal fired, draft at the flue gas outlet, flue gas temperature, rate of burning, and smoke density. In addition, the magazine cover was opened momentarily at 15 min intervals for observation of fuel bed reactions.

A continuous record of smoke density at the flue gas outlet was obtained by use of a recorder of the photoelectric cell type. This instrument has been described in the literature [1] and furnishes a very accurate means of measuring smoke densities.

As in the anthracite tests, a preliminary starting fire was used. During this preliminary period the draft at the flue gas outlet was maintained at 0.05 in. water, and the primary air opening was in the *open* position.

At the end of the preliminary period a full charge of coal was placed in the heater and the test started. The test was of 6 hours duration, divided into three periods. *1st and 2nd hours:* Draft 0.05 in. water—primary air open; *3rd and 4th hours:* Draft 0.025 in water—primary air closed; *5th and 6th hours:* Draft 0.150 in. water—primary air open.

Conditions during the first period of the test were approximately those at which nominal rated output would be achieved. The second period simulated banked operation, and the third period may be considered a drive rate. The marked changes in operating conditions between the periods furnished an opportunity to observe the reaction of the heaters to *check-down* and *pick-up* settings.

No flue gas limitation restrictions were imposed, and the results are directly the effect of draft variation. Where independent means were provided, the secondary air inlet was left in full open position.

TEST RESULTS

Anthracite Tests: The anthracite used in all tests was from No. 6, D, L, and W seam, nut size, and of the following analysis:

Proximate analysis, as fired	Per Cent
Moisture	1.5
Volatile matter	4.1
Fixed carbon	83.6
Ash	10.8
Calorific value, as fired, Btu per lb—12,880	
Carbon	76.6
Hydrogen	4.1

The leading test information is given in Table 1, and the heat balances are presented graphically in Fig. 2. A study of these data shows a large variation in the efficiencies and outputs obtained with the various heaters.

[1] Bituminous Coal Research, Inc., Technical Report No. 1, Part II, Burning Tests on Four Typical Coals, by R. A. Sherman, E. R. Kaiser, and H. R. Limbacher, Appendix A, p. 43-47.

Heater A had an output of 47,688 Btu per hour at an efficiency of 70.4 per cent. This heater is designed with a liberal combustion space, and has considerable heating surface in contact with the gases of combustion. The advantage of this design is reflected in the lower flue gas temperature and dry flue gas loss.

The higher flue gas temperature obtained with Heater B is contributory to the high dry flue gas loss in the heat balance. This heater was deficient in heating surface.

Although Heater C was of similar external size, the magazine capacity was considerably less than that of the other heaters. This resulted in the shorter length of test, as all tests were conducted with the magazine filled

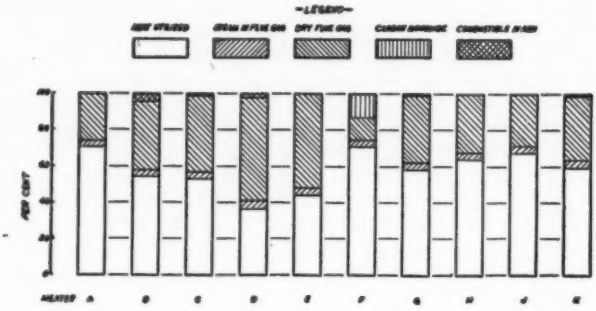

FIG. 2. GRAPHICAL HEAT BALANCE DATA—ANTHRACITE TESTS

to the normal top level. The high dry flue gas loss in this test was due in the most part to the low CO_2 and the high excess air. It was apparent that the arrangement of the secondary air tuyeres in this heater was not proper.

Heater D exhibited this same fault to an even greater extent, the excess air being very high. Heaters C and D both took the secondary air from a space common with the primary air. The desirability of separate control of secondary air is noted.

The test with Heater E gave a low CO_2 and high excess air, although separate control of secondary air was provided. As a supplement to the test of the heater as manufactured, the area of the secondary air ports was reduced, and a second test run, with improved results, as shown in the following brief summary:

HEATER E	AS MANUFACTURED	MODIFIED
CO_2—per cent	5.3	6.9
Excess air—per cent	262.0	171.0
Fuel burned—lb/hr.	4.81	5.18
Overall efficiency—per cent	44.0	54.7
Output—Btu/hr.	27,259	36,495

Heater F was of somewhat larger size. This is reflected in the larger fuel charge and the longer firing period. This heater was so designed that it could be operated with either overfeed or underfeed burning in inverted position. For comparative purposes, the underfeed method was employed in the anthracite test.

The CO_2 obtained in this test was good, and due to an abundance of heating surface, the flue gas temperature was low. This resulted in a low

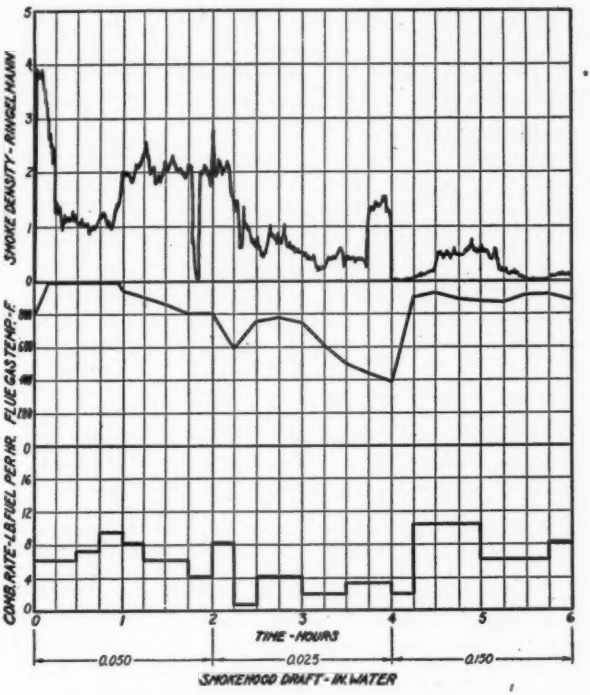

FIG. 3. HEATER A—GRAPHICAL TEST LOG

dry flue gas loss, but the loss due to the formation of CO was very high, and to an extent nullified the dry flue gas effect. The deep fuel bed of the magazine feed heater does not lend itself well to underfeed combustion, as the CO_2 formed in the first section of the bed is largely reduced to CO in the latter stages of combustion. Periodic gas analyses made during the test showed CO percentage as high as 5.8.

This heater was not provided with any means for providing secondary air and, unfortunately, it was not possible to make any changes to try the

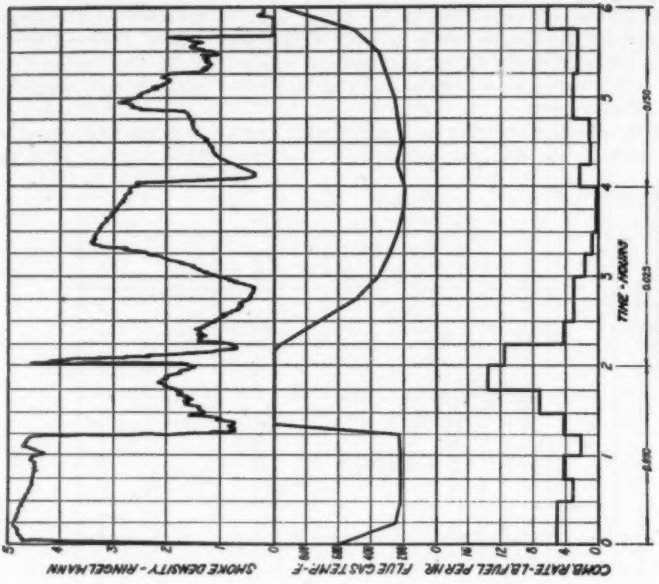

FIG. 5. HEATER C—GRAPHICAL TEST LOG

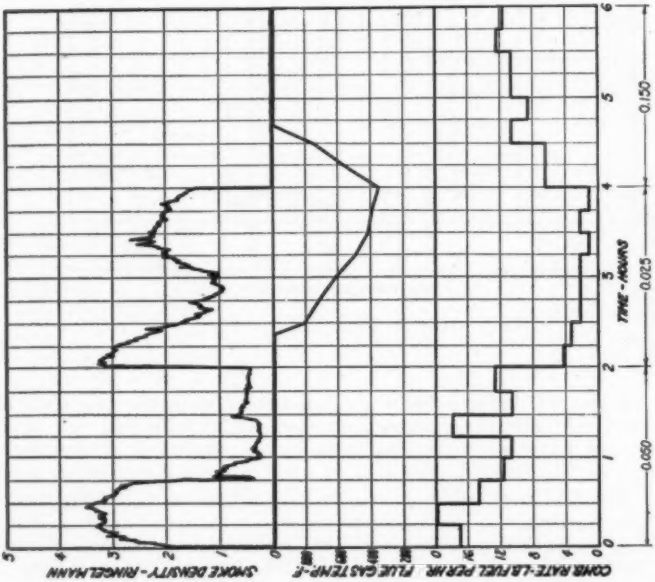

FIG. 4. HEATER B—GRAPHICAL TEST LOG

effect of such provision. Despite the high *CO* loss, this heater showed an efficiency of 70.8 per cent at an output of 58,088 Btu per hour.

Heater G was equipped with a separately controlled secondary air inlet. Good combustion was attained, but the heater is markedly deficient in heating surface, and the high flue gas temperature resulted in a correspondingly high dry flue gas loss.

The flue gas temperature obtained in the test with Heater H was relatively low, and as a result, the lower dry flue gas loss resulted in an efficiency of 64.1 per cent at an output of 43,923 Btu per hour. Although Heater J, employing the cross-feed principle, showed a relatively high CO_2 in the flue gases, the lack of heating surface and short flue travel resulted in a high dry flue gas loss. The high CO_2 value and lack of measurable quantity of *CO* indicates proper distribution and utilization of combustion air. The performance of Heater K was characterized by a high dry flue gas loss with attendant low efficiency.

A general review of the tests conducted with anthracite leads to the following conclusions:

Although the heaters, with the exception of Heater F, were all of similar external size, there is considerable variation in the output, and efficiency values.

Heaters A and F were provided with a reasonable amount of secondary heating surface, and flue travel was provided between the combustion chamber and the flue gas outlet. The advantage of this construction over the other heaters, in which the gases passed directly from the combustion chamber into the flue gas outlet, is clearly indicated by the efficiencies and outputs obtained.

The disadvantage of faulty application of secondary air, with its attendant high excess air, is evidenced by lower efficiencies, and outputs. This is particularly true of Heaters C, D, E, and K.

The anthracite test conditions may be considered as applicable for determining the rated output of the heaters. It is pointed out, however, that although the heaters are capable of delivering the respective outputs, it does not follow that in all cases satisfactory heating of a space having a similar heat loss will be obtained. These heaters are of the radiant type and, where used to provide heat for more than one room, much depends upon the circulation of the air from room to room as governed by the heater location, and construction of the house.

Bituminous Tests: The bituminous coal used in these tests was from the Island Creek seam, 2 in. x 3 in. size, and of the following analysis:

Proximate analysis, as fired	Per Cent
Moisture	1.8
Volatile matter	38.1
Fixed carbon	55.5
Ash	4.6

The principal objective of this test series was to determine the relative smoke-producing tendencies of the several heaters. The Island Creek coal was selected for the test program because of its high volatile content. Rose and Lasseter [2] have noted the following:

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[2] Smoke-Producing Tendencies in Coals of Various Ranks, by H. J. Rose and F. P. Lasseter. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 329.)

As would be expected, high-volatile bituminous coals produced the most smoke and soot . . . others have reported that smoke production is proportional to the *volatile matter* of coal, which may well be the case with a limited range of coals, particularly of the bituminous class.

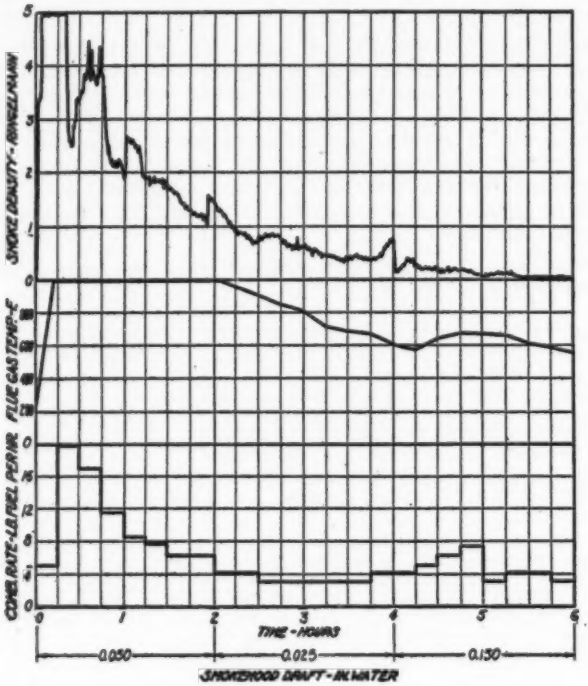

FIG. 6. HEATER D—GRAPHICAL TEST LOG

The results of the bituminous tests are presented in a series of graphs, Figs. 3 through 12. The flue gas temperature and the rate of burning are shown in addition to the smoke record.

As shown in Fig. 3, the average smoke density for Heater A was 1+ according to the Ringelmann Chart. Following a momentary peak of No. 4 density, immediately after firing, the smoke decreased sharply. A short peak is shown at the start of the bank-down period. The rate of burning did not decrease as rapidly as the air supply, with a resultant deficiency of air evidenced by the smoke peak. This is analogous to the momentary smoke peak found at the start of *off* period operation of the conventional under-feed bituminous coal stoker. This smoke peak at the start of the bank-down period is characteristic of all the heaters tested.

The flue gas temperature exceeded 1000 F (the maximum of the temperature measuring instrument) for about 30 min in the first section of the test. The flue gas temperature and rate of burning reacted rather slowly to checking of the fire, but an immediate response to pick-up is shown at the

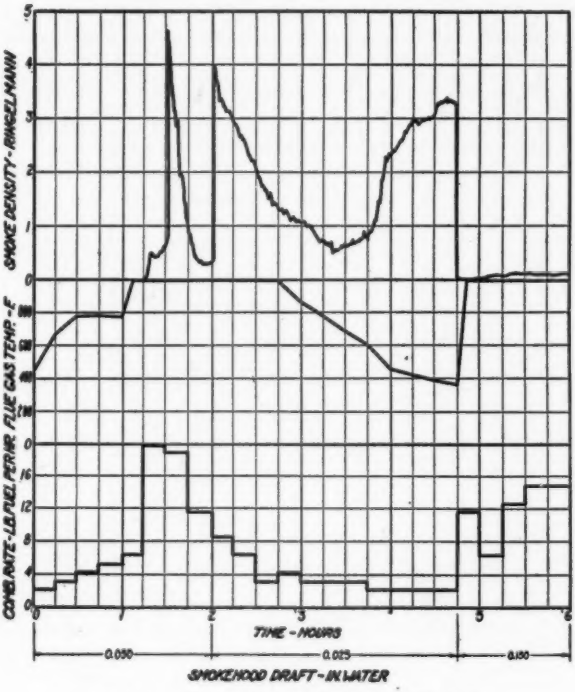

FIG. 7. HEATER E—GRAPHICAL TEST LOG

start of the third period. A slight amount of smoke-back from the upper magazine was found when the charging door was opened during the bank period. No top reaction was noted in the other periods.

Fig. 4 shows the test results for Heater B. The smoke peaks in the first and second periods of the test were of fairly long duration. No smoke was produced in the drive-rate period, however, and the average density for the entire test was 1+ Ringelmann. The absence of smoke in the last period is attributable to the rather complete coking of the coal in the previous periods and the dilution of the smoke at high excess air.

The flue gas temperature was above 1000 F throughout the entire first period, and the fuel burned at high rates. The heater reacted sharply to

change of damper setting at the start of the bank-down and pick-up periods. Some smoke-back was noted when the magazine cover was opened during the first and second periods. Because most of the volatile had been released before the third period, no smoke-back was found in this period.

The smoke record for Heater C, shown in Fig. 5, indicates a long period, approximately 70 min, of dense smoke at the start of the test. The dense

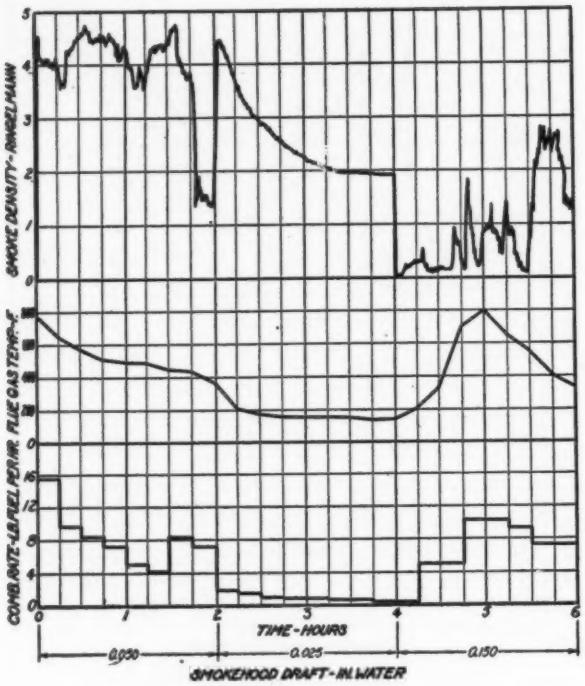

FIG. 8. HEATER F—GRAPHICAL TEST LOG

smoke persisted until ignition of the gases above the fuel bed was established. This is evidenced also by the temperature record. The average smoke density of 2+ Ringelmann indicates the lack of proper air supply in this heater.

The flue gas temperature and rate of fuel burning were very low until top ignition of the gases occurred. The heater responded quickly to the decrease in air supply at the start of the second period, but a very poor response to pick-up is shown in the third period. Despite the dense smoke produced by this heater in the first period, little smoke-back occurred when the magazine top was opened.

The smoke record for Heater D, Fig. 6, shows an initial peak of smoke of No. 5 Ringelmann at the start of the test. This peak tapered off steadily until the end of the first period. Smoke of less than No. 1 density persisted throughout the remainder of the test. The high rate of burning during the first period resulted in abnormally high flue gas temperatures at this time. The rate of burning did not show a marked decrease at the beginning of the bank period, as a downward trend was already well established. This was brought about by the increasing compactness of the fuel bed caused by the caking and coking of the fuel. The heater responded moderately well at the start of the pick-up period.

No smoke-back or flare was found on opening the magazine cover except at the start of the test, when strong smoke-back was noted. Ignition over the fuel bed occurred at 11 min after charging the magazine. An explosion of strong intensity was noted when this ignition was established.

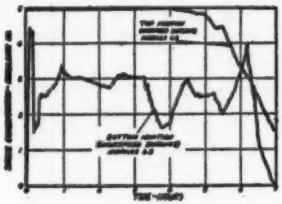

FIG. 9. COMPARATIVE SMOKE RECORD (HEATER F)

Fig. 7 illustrates the results of test with Heater E. No smoke record was obtained in the early stages of the first period, but observation of later record indicates the probability of excessively dense smoke. The smoke peak at the start of the bank-down period was very sharp, and it is shown that the rate of burning immediately preceding this period was high. The bank period was conducted for 2¾ hours instead of 2 hours used in other tests.

The smoke dropped sharply to zero at the start of the pick-up period and the rate of burning increased rapidly. This is further evidenced by the immediate rise in the flue gas temperature. Some smoke-back was found in the first period, but subsequent ignition from the active fuel bed stopped this effect.

The data for Heater F are depicted in Fig. 8. Despite the principle of inverted underfeed burning employed, this heater smoked badly throughout most of the test period. It is noted, however, that no provision was made for secondary air. The lower flue gas temperatures and relatively low rates of burning are evidence of the effect of the fuel bed resistance to the flow of air for combustion. This is particularly noticeable during the bank period when very low rates of combustion were attained. It is noteworthy, also, that the response to pick-up conditions was relatively slow.

This test was characterized further by persistent smoke-back when the magazine cover was opened. This condition was very bad during the pick-up

period. Examination of the fuel bed at the end of the test showed very strong coking and bridging, and the fuel bed was burned out in the lower section, leaving a void above the grate.

It has been stated previously that Heater F could be operated on either the under-feed or over-feed principle. As a matter of interest, trials were made of both methods. Illinois No. 6 coal, 2 in. x 3 in. size, and having a volatile

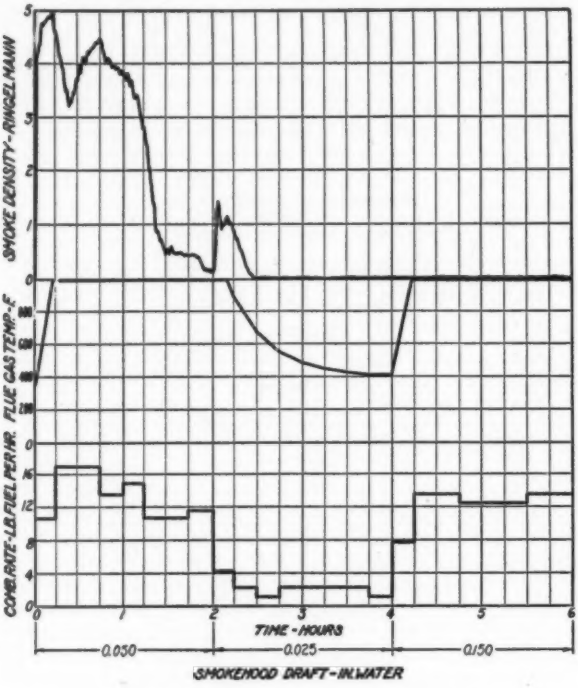

FIG. 10. HEATER G—GRAPHICAL TEST LOG

matter content of 33.9 per cent, was used in these tests. This free-burning fuel was selected in an effort to equalize fuel bed conditions by the reduction of the coking and bridging effects. Tests were run at a draft of 0.06 in. water at the flue gas outlet. The flue gas temperature was controlled by adjustment of the primary air damper so that a value of 900 F was not exceeded.

The comparative smoke record, shown in Fig. 9, clearly demonstrates the advantage of underfeed burning. Although excessive smoke was produced with the underfeed method in the regular test, it is evident that the smoke would have been much greater if the overfeed principle had been used.

The smoke record for Heater G, shown in Fig. 10, shows a short period of dense smoke at the start of the test. The smoke diminished fairly rapidly and except for the characteristic lesser peak at the start of the bank period, no more smoke was produced. The method of introducing secondary air was quite effective in reducing the smoke emitted. The heater was very responsive to draft and damper setting changes, as shown by the sharp

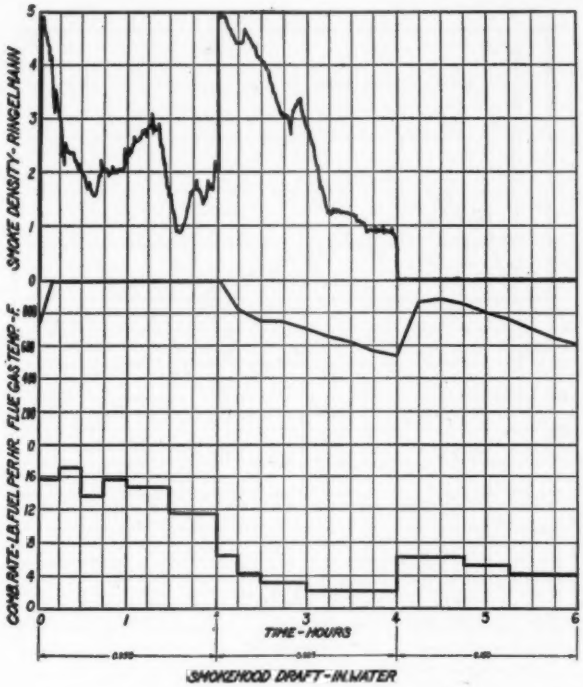

FIG. 11. HEATER J—GRAPHICAL TEST LOG

trends of temperature and rate of combustion at the start of the bank and pick-up periods. No smoke-back was found at any time in the test. Ignition of the gases above the fuel bed was started early in the test and was maintained.

It was not possible to test Heater H for its smoke-producing tendencies, and testing of this heater was restricted to the anthracite run previously reported.

Fig. 11 illustrates the test data obtained with Heater J. This heater showed a characteristic peak of dense smoke at the time of charging the fuel and at the start of the bank period, but the smoke density diminished fairly

rapidly to acceptable limits. As shown by flue gas temperature and burning rate, the heater reacted positively to draft variations.

Some *puffing* was noted during the test and a considerable amount of smoke and flame was emitted from the charging door when this was opened at the regular inspection intervals. This reaction was particularly in evidence during the first two periods of the test, when the volatile matter was being released from the fuel bed.

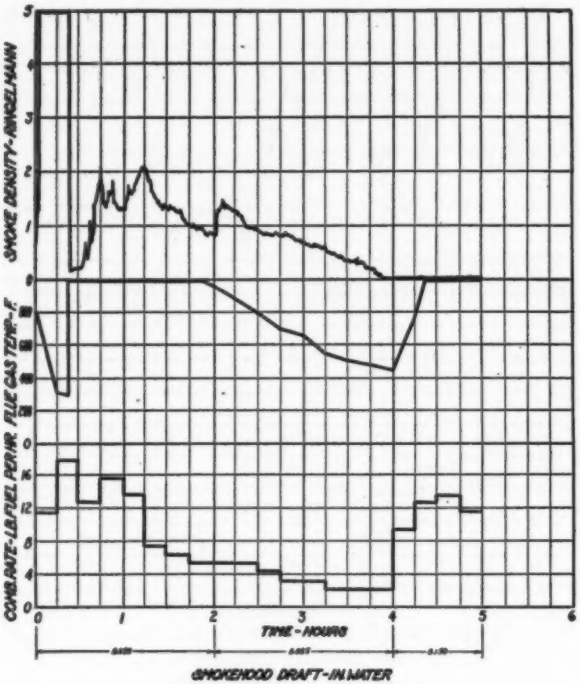

FIG. 12. HEATER K—GRAPHICAL TEST LOG

Test data for Heater K are shown in Fig. 12. Following the initial peak of dense (No. 5 Ringelmann) smoke, the density decreased to a moderate value throughout the remainder of the test. This was due largely to dilution, as the anthracite rating test for this heater had shown very high excess air.

The heater responded immediately, and rather sharply, to variations in draft. This is shown by the decided fluctuations in flue gas temperature and rate of burning when the test control conditions were changed. Several moderate *puffs* and a slight amount of smoke-back were noted during the first period of the test.

Banking Tests: In addition to a quick response to changes in draft and damper settings, a properly designed heater should be capable of maintaining a uniform, low rate of combustion when operating under banked conditions. This is essential in order to prevent overheating in mild weather and to insure satisfactory extended periods of low output operation. Bureau of Standards, TS-3443 states:

". . . the heater shall maintain a fire at an average fuel consumption rate of not more than 25 per cent of that required for the maximum rating for a minimum of 24 hours for magazine feed heaters."

Several of the heaters included in this program were tested under banked conditions and the following general conclusions were made: It is essential that draft doors or dampers be tightly fitting, preferably with ground joint construction; and secondary air should not be admitted in a space common

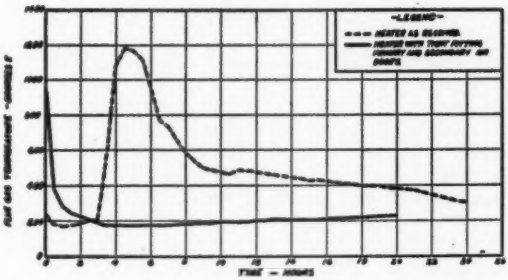

FIG. 13. BANKING TEST PERFORMANCE (HEATER K)

with primary air, because with certain fuel bed characteristics it may function as primary air, thus preventing a proper control of rate of burning.

Fig. 13 illustrates typical results of a banking test with a high-volatile free-burning bituminous coal, showing the effect of proper and improper control of the rate of burning. The smokehood draft was maintained at .0.06 in. water, simulating cold weather operation with a reasonably effective chimney.

CONCLUSIONS

Inasmuch as the heaters used in this study were each of a different design, and three different types of fuel burning were encountered, it may be said that the results are typical of the performance of coal-fired magazine space-heaters now available. No brief is offered herein as to the relative merits or demerits of any particular make of heater, but an attempt has been made to point out the advantages or disadvantages of basic principles.

Future developments may lead to a different general situation but, for the present, this study may be found of assistance in judging the performance expectations of coal-fired space heaters of the magazine type.

DISCUSSION

S. K. SMITH, Westfield, Mass.: Will the author explain his reaction and comments on the high *CO* obtained in one of the tests. I believe it was the test that had the highest efficiency.

MR. CROSS: In general, the trend was the other way. We experienced low CO_2 very high O_2, and negligible *CO*, except for the underfired heater, which had an appreciable amount of *CO* in the early period. These heaters are rather loose in their construction and the tendency is to very high percentages of excess air.

MR. SMITH: It answers the question, but I was looking at Heater F. That did show, if I read it correctly, an appreciable amount of *CO*—a large amount.

MR. CROSS: You are looking at the heat balances in the anthracite test.

MR. SMITH: That is right.

MR. CROSS: The principal reason for the high *CO* obtained in the test of Heater H was that this heater had no provision for the admission of secondary air.

1284

No. 1284

THE APPLICATION OF HIGH TEMPERATURE LIQUID AS A HEAT OR ENERGY TRANSFER MEDIUM FOR DOMESTIC AND INDUSTRIAL USES

By ORION O. OAKS,* RARITAN, N. J.

THE use of high boiling point liquids as heating media is not new, but the liquids formerly used, however, were limited in regard to maximum or minimum workable temperatures, and in many industrial applications were vaporized with the vapor being used under various pressures to obtain the desired temperatures.

The author intended to use such a liquid to obtain high temperatures without pressure and to employ the heat of the liquid only. The first experiments in 1935 were unsatisfactory due to the fact that any liquid, then obtainable by the author, having a boiling point high enough for the desired purpose had a freezing or solidifying temperature too high for practical use, especially as it was desired to use the material in the primary or exclusive heat or energy generator. The properties of the liquid now being used enabled the author to resume experiments which are described in this paper. The apparatus as developed and used is the invention of the author.

In the domestic field it is proposed to provide all of the heat and mechanical services from a single heat source and to employ a liquid as the heat or energy transfer medium to operate the various services such as house heating, domestic hot water supply, cooking, baking, refrigerating, drying, air conditioning, and generating electrical energy for light and power. This procedure is in contrast with the conventional services which may involve three or four types of heating equipment operating at various efficiencies.

By burning any preferred type of fuel, such as coal, gas, or oil, in a single furnace having a high efficiency and distributing the heat by means of the circulated heated liquid to the various equipment units where it is utilized at 100 per cent efficiency, the losses are limited to those of the boiler or heater

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and the radiation from the transmission lines and equipment units. The efficiency of the new types of insulation, properly applied, reduces the radiation and convection losses to a small per cent, and if the efficiency of the boiler is high, the over-all performance becomes highly satisfactory.

The scope of the experiments described herein was limited to demonstrating the feasibility and practicability of the apparatus, to determine the level of convenience and ease of operation to the householder or building operator, and

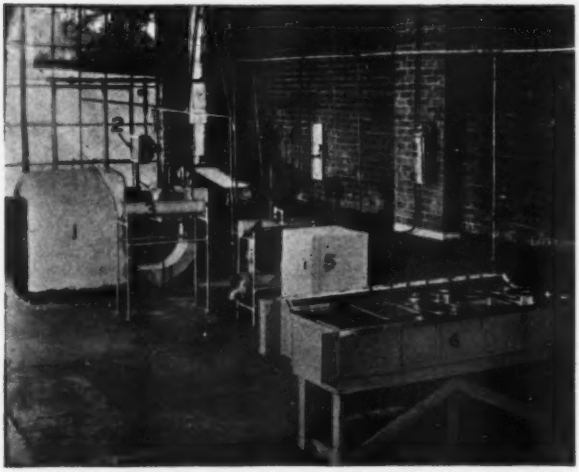

FIG. 1. EXPERIMENTAL INSTALLATION FOR TEST OF HIGH TEMPERATURE LIQUID AS HEATING MEDIUM

Legend

1. Boiler
2. Unit Heater
3. Hot Water Heater
4. Steam Generator
5. Refrigerator
6. Cooking Unit

to exploring the extent of its application to commercial use and to industrial and manufacturing processes.

Tests of a practical type were conducted to determine the economy of the operation and the cost compared with that of conventional equipment in current use. The method employed involves the circulation of hot liquid from the primary generator or boiler to and through jacketed baking ovens, submerged coils, radiant panels or coils, and the return of the liquid to the boiler for reheating and recirculating.

An experimental installation including (1) boiler, (2) unit heater, (3) hot water heater, (4) steam generator, (5) refrigerator, and (6) cooking unit, is shown in Fig. 1. The ovens attached to the same installation are shown in Fig. 2. Various parts of the equipment are shown as follows: Fig. 3, Boiler;

Fig. 4, Expansion tank; Fig. 5, Cooking pot and recess, and Fig. 6, Upper and lower oven units. The liquid circuit is shown on the flow diagram, Fig. 7.

PRIMARY GENERATOR

The boiler or primary generator is of the liquid-tube, liquid-wall type with a built-in economizer and a built-in air preheater. These auxiliaries are desirable to maintain the high efficiency required of the generator because, without

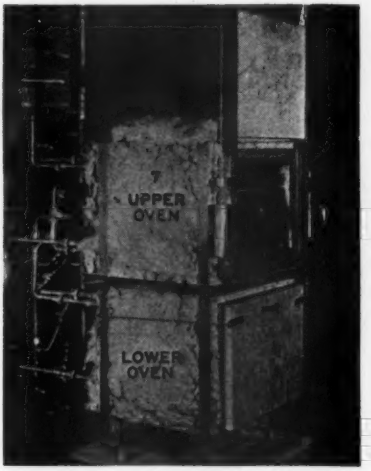

FIG. 2. OVEN UNITS USED WITH EXPERIMENTAL INSTALLATION OF FIG. 1

them, the high temperature of the heated liquid results in a high temperature of the products of combustion.

The heat exchange effected in the economizer and air preheater reduces the stack gas temperature sufficiently to make possible an overall efficiency well above 80 per cent.

SERVICE UNITS

The *house heating unit* may be of the liquid-tube, air-heating type with all the necessary and customary features, such as humidifier, air filter, circulating fan and, if desired, a cooling and dehumidifying coil for summer cooling. This unit, which is under complete automatic control, is contained in a complete housing and may be located at any convenient point in the building. Hot water radiators, steam radiators, convectors, or radiant heating are a few of the means by which heating may be accomplished. Radiation and convection

from *direct* circulation of the liquid through any desirable form of heating surface could also be used for heating.

The *hot water supply unit* is in the form of a heat exchanger with submerged hot liquid coils and may be of the tankless, instantaneous type or may be used in connection with a storage tank.

The *refrigerating unit* is a standard machine of the absorption type and is adapted to use with liquid heat by the simple substitution of a hot liquid coil in place of the existing heat source.

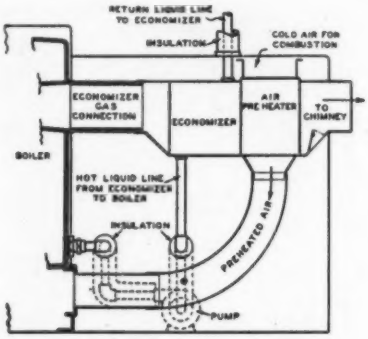

FIG. 3. BOILER UNIT

The *cooking unit* consists of metal pots of various sizes which are depressed in the insulated working table. Fitted to these metal pots are cooking utensils or liners which are provided with insulated metal covers. Set flush with the work table are a service hot plate and a flat surface griddle to be used for all surface cooking and frying. A small sink is incorporated in the work table for convenience in food preparation. For filling the utensils while in place, a combination, movable, trigger-acting faucet forms part of this equipment.

The *baking unit,* which may be located separately or built into the work table, consists of a compartment, or compartments, conforming to standard oven dimensions and is provided with *raise-up* or *open-out* flush mounted doors as desired.

The grilling of steaks, chops, or other meats is accomplished by placing them in close proximity to the ceiling of the oven in which position results are obtained identical with, if not superior to, any other methods in current use.

Due apparently to the exclusive use of *radiant heat,* the foods are very uniformly cooked and possess a flavor and texture superior to foods cooked by other methods. This statement is made solely upon the basis of comments of laymen and experts who have eaten the food cooked by liquid heat. In general, the time required for cooking is approximately 50 per cent of that required by conventional methods.

The ovens used in the experimental plant are not ventilated, and all experiments to date indicate that no ventilation is necessary or desirable.

The ovens when suitably equipped can be utilized as a dish washer, thereby utilizing the heated surface for a rapid production of steam and boiling water which is sprayed over the dishes. After rinsing, the oven is used as an effective drying device.

The toaster consists of a jacketed tunnel, a cross section of which is approximately the shape of a slice of bread. The degree of the toasting is governed by the speed at which the bread is conveyed through the tunnel.

The time required for raising the various units from a cold condition to a working temperature varies from 30 seconds in the case of the cooking pots, griddle, service plate, and toaster, to 2 min in the case of the ovens.

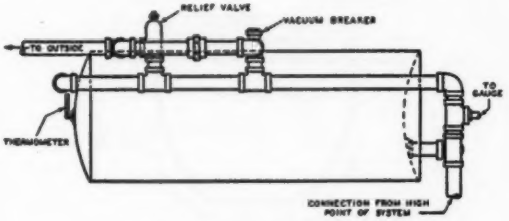

FIG. 4. EXPANSION TANK

The drying of laundry and the dehydration of fruits and other foods are readily accomplished by radiant heat and/or convection as required. The use of liquid heat coils or surfaces controlled at the appropriate temperature and air supply is employed as, and when, required. The size and number of the previously described units may be varied from requirements for a very small home to those of apartment houses, restaurants, hotels, hospitals, or a great institution.

EXPANSION TANK

The hot liquid circulates through the system at atmospheric pressure; therefore, an expansion tank is used to absorb the increasing volume of the liquid as the temperature increases. This vessel is cylindrical in form and is provided with an inlet connection from the high point of the system and an overflow pipe which discharges to the atmosphere through a low pressure relief valve. A liquid gage is employed to indicate the height of the liquid in the system.

CIRCULATING PUMP

A motor driven centrifugal pump is used to circulate the liquid throughout the various units and return it to the generator for reheating and recirculation.

This pump operates only against the head created by frictional resistance through the lines.

ELECTRICAL ENERGY

When required, electric energy is produced by means of a heat exchanger or boiler producing superheated steam at pressures varying from atmospheric to 750 psi. This steam is utilized in driving a steam turbine connected to a generator to produce electric current of whatever characteristics may be required. This turbo electric plant may be operated under various conditions, depending upon the presence of uses for process steam, availability of suffi-

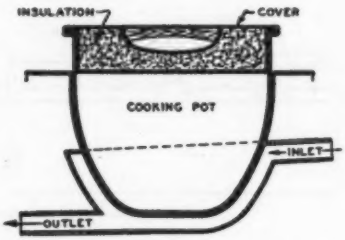

FIG. 5. COOKING POT AND RECESS

cient condensing water, or by any other factors governing power plant uses. The electric generator plant may be regarded and used as the major source of electric energy or may be considered as an auxiliary standby unit for use in the event of current failure from other sources. For domestic use the steam turbine may be operated at pressures not exceeding 15 psi in order to take full advantage of the outstanding safety and hazard-free features of the entire apparatus. In this connection it is to be noted that the liquid circuit is operated at practically no pressure throughout its entire range except that caused by its own static head.

LIQUID DATA

The following incomplete data may be given in connection with the liquid employed as the heat distributor or heat vehicle: Name—Tetracresylsilicate; Boiling Point—817-825 F; Solidifying Point—Undetermined, though liquid viscous at —65 F; Specific Gravity—1.13; Specific Heat—0.43; Nonpoisonous; Noncorrosive; Nonexplosive; Nontoxic; Viscosity similar to *SAE* 20 at 70 F; Extremely fluid at temperatures above 100 F.

The heat transfer properties of the liquid follow closely those of other high boiling point liquids and are similar to those of water at like temperatures, namely, 300 to 600 F, which is the range that is considered in this paper.

TEST RESULTS

The amount of liquid circulated through the various equipment units was calculated by establishing the amount of liquid to be circulated per 1000 Btu

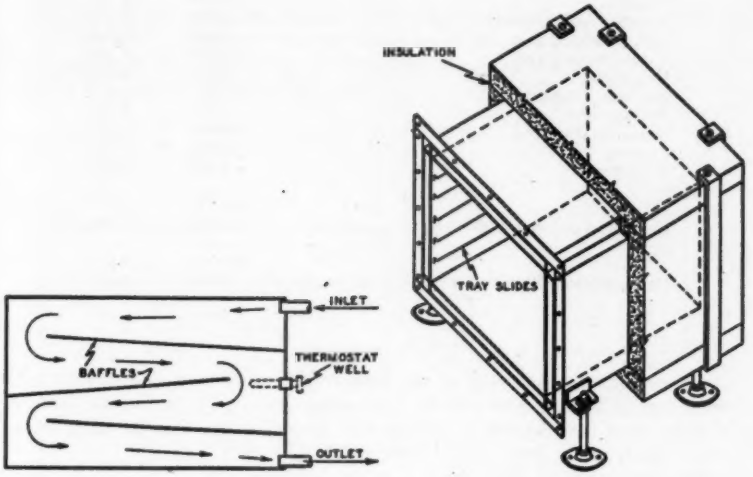

FIG. 6. GENERAL ASSEMBLY OF INSULATED OVEN

output and computing the total liquid circulated at an assumed temperature drop 600 − 400, or 200 F.

$$1 \text{ gal liquid} = 8.3 \times 1.13 = 9.4 \text{ lb}$$
$$9.4 \times 200 \times 0.43 = 800 \text{ Btu per hour per gallon}$$
$$\frac{1000}{800} = 1.25 \text{ gal circulated per hour}$$

Then the liquid may be expressed as gpm, or $\frac{1.25}{60} = 0.02085$ gpm, or 0.021 gpm per 1000 Btu required.

The pump was set to operate at 5 gpm, and at this rate the temperature drop was less than that calculated. The temperature drop actually was 125 deg with all units functioning and, of course, became less as part of the units were turned off. This would indicate that certain units consumed less heat than expected or that the pump was handling more liquid than calculated.

The cooking unit apparently used less heat than expected because the drop through this unit was only 15 to 25 F.

The pilot plant requirements were as follows:

	Btu	gpm
Griddle	11354	0.237
3 small cooking pots	12000	0.252
Large cooking pot	8000	0.168
Large cooking pot	8000	0.168
Hot plate	4800	0.1005
Hot water supply	20750	0.435
Refrigerator	3000	0.063
House heating coil	60000	1.260
2 ovens	24000	0.505
1 toaster	1300	0.0295
	153204	3.218
Light and power (5000 watts)	42500	0.893
	195704	4.111
Radiation loss 10 per cent (Piping not insulated)	19570	0.411
	215274	4.522

An approximate expression of the performance of a blast coil was obtained from a test of a small heater made from copper tubing ⅜ in. outside diameter and three rows deep having a housing and an outlet face area 13 in. x 10 in., or 0.905 sq ft and a total heating surface of 6.76 sq ft.

Data from a test of this heater were as follows:

Outlet area	0.905 sq ft
Velocity of leaving air	600 fpm
Entering air temperature	80 F
Leaving air temperature	200 F
Volume of leaving air	543 cu ft
Entering liquid temperature	550 F
Leaving liquid temperature	475 F

Without correction of air volume for deviation from standard air, the heat added to the air would be $543 \times 0.075 \times 0.24 \times 120 \times 60 = 70{,}000$ Btu. The approximate heat per square foot of heating surface is 10350 Btu per (hr) (sq ft), and the heat emission is approximately 32.1 Btu per (sq ft) (deg difference in temperature between average air and the average liquid). Several other supporting tests showed like results when the temperature conditions were near to those given in the foregoing.

The ovens presented difficulties in making any accurate tests, but it was found that a metallic, unshielded thermometer placed in the center of the interior volume gave a reading not more than 2 or 3 deg below the average temperature of the circulating liquid.

The ovens were unvented except for a small amount of leakage around the doors. The steady-state oven temperatures were reached within 5 min after

turning on the circulating liquid and were practically uniform throughout the entire oven volume. Two ovens were constructed and mounted, one above the other, and the temperatures of both were alike at all times.

CONTROLS

The problem of automatic controls presented requirements which could not be satisfied in the present extensive offerings of the control industry. It was necessary, therefore, to develop suitable control equipment in the experimental plant as completely as all other features. All units are automatically con-

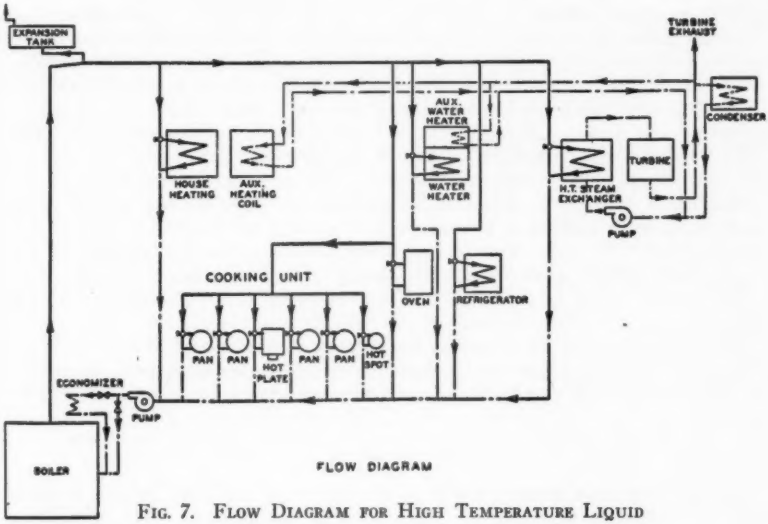

FIG. 7. FLOW DIAGRAM FOR HIGH TEMPERATURE LIQUID

trolled by a standardized piece of equipment combining a three-way modulating liquid valve with a liquid-filled expansion system which serves to bypass all or any part of the flowing liquid around any unit in response to rising temperature, and conversely, to increase the supply of hot liquid through the unit upon falling temperature. Means of adjustment of the operating point and the control differential are provided so that the operator may vary the temperature of the unit at will or turn it off completely.

After setting this automatic control, the temperature of the unit is held within close limits as long as the flow of the hot liquid continues. The temperature of the hot liquid circuit from the boiler is maintained constant at any desired point. The boiler control operating the fire or the firing device is responsive to the temperature of the outgoing liquid. By this procedure

any unit may be operated at a different temperature from any other unit or may be turned off completely.

ECONOMICS

The materials used in the construction of the apparatus may be selected from a wide range of types, as they are limited only by their ability to withstand heat within the temperature range. The boiler, piping system and the various units may be constructed of either ferrous or nonferrous materials, as experimental data collected to date indicate no action by the liquid upon any metal other than that due to temperature. Joints may be either welded, screwed, flanged, or brazed. Therefore, it is apparent that the kind of material used in the piping system will be influenced by the cost of fabrication.

The exposed portion of the units such as pots, griddles, ovens, and so forth, should be constructed of a material such as stainless steel which presents a fine appearance, does not readily oxidize under high temperature, and possesses great resistance to the stresses set up by the wide range of temperatures involved. The component parts of the apparatus are subject to great flexibility regarding the relative location and may be placed according to the requirements of any type of structure, but the cost will be reduced greatly by the consolidation and streamlining of the various units so that the benefits of the prefabrication may be obtained and the cost of field construction may be reduced to a minimum. The cost of the liquid is not a serious matter, as it is more or less a permanent part of the system, does not dissipate or evaporate, and requires only very slight replenishment over long periods of time.

The underlying reason for possible savings when using the liquid heat consolidated unit is the fact that all energy conversions are 100 per cent efficient excepting in the primary generator. By concentrating on the efficiency of this primary generator and reducing the loss through radiation by utilizing improved types of insulating materials, transmission losses are reduced to a minimum, and the over-all efficiency of the entire equipment is raised to a material extent.

CONCLUSION

Since a full scale working model of the apparatus has been under successful test operation and study for the past year, it is concluded that for the various domestic service requirements the practicability of using a high temperature liquid of the type described in the equipment outlined has been demonstrated.

DISCUSSION

P. A. GREEN,[1] Philadelphia, Pa.: In the second slide you showed a unit heater of 70,000 Btu capacity. Could you tell me what size fan, or how many cfm the unit handled, and what type of radiator was used?

MR. OAKS: I did not state that because it is given quite in detail in the paper. The outlet size of the heater is 10 x 13 in. It is constructed of 3/8 in. diameter plain

[1] John J. Nesbitt, Inc.

copper tubing, three rows deep. I believe it has a total heating surface of 7.6 sq ft. The output was probably about 550 cfm, with 600 ft velocity. Face velocity, entering temperature of the air and the liquid and outgoing temperature of the air are also given in the paper. The Btu output per square foot of heating surface is also given. You will find all that in the paper.

R. L. STINARD, New York, N. Y.: What is the relative cost between the heating system you propose and a conventional steam or hot water system for residence use?

MR. OAKS: I did not include that phase of it in the paper because it was somewhat controversial. However, figures that I have made and all the data that I have been able to compile would lead to the conclusion that the apparatus can be built for at least no greater cost than the present apparatus used for the services included in a home under present-day conditions. The cost of operation—again I give this verbally until I can completely substantiate it—will probably be 50 per cent less than the cost of operation of conventional materials and appliances.

H. M. NOBIS, Cleveland, Ohio: What is the cost of the liquid, and how is it sold?

MR. OAKS: Right now the liquid is in the laboratory pilot plant stage. It costs slightly more than it will when it is made in quantity. According to our manufacturer it may be sold as low as 50 cents per gallon, and the amount used will be very small. I do not believe the householder will use over a gallon every five years.

MR. NOBIS: Did the liquid remain stable during the operation?

MR. OAKS: All I can say is that the liquid in the pilot plant has been in operation for two years. I have had it re-analyzed and the chemical engineers tell me that for the purpose intended, it is even better now than the day we first started to use it. It has been in and out and in and out, exposed to the air, in barrels and on the floor, subjected to every imaginable abuse, and it still seems to operate, as far as I can see, exactly the same as it did the first day.

B. M. BROD, New York, N. Y.: What are the possibilities of using this liquid as a substitute for brine in low temperature refrigeration plants?

MR. OAKS: Well, as you realize, its specific heat is 0.43, which means that about twice as much liquid would have to be circulated to produce the same results. The liquid has a viscosity similar to *S.A.E.* 20 oil at 70 F, although it is very fluid at 100 F. If a low temperature were used, we would not only be up against a specific heat which might be too low, although it could be overcome by a higher pumping rate, and a greater flow resistance due to the viscosity of the liquid at low temperature. It can, however, be pumped at −65 F.

H. F. BRINEN, Racine, Wis.: Was there any difficulty in holding this liquid in ordinary screw type or flanged joints? Does the liquid have an ordinary surface tension, or, like many liquids of this type, is it low?

MR. OAKS: All types of piping were included in my pilot plant, iron, brass, copper, screwed joints, welded joints, soldered joints—although, of course, we had to use silver solder. I can only say that it is necessary for the constructor to do a good job. If he has a tight joint, he may use any of the conventional types of construction. The surface tension in the upper levels is less than some other liquids. After we learned, however, to make good joints we had no trouble whatsoever from leakage of the piping system and I do not think leakage would be a material feature.

L. L. GAREY,[2] Newark, N. J.: How close is this experimental equipment to commercial production?

MR. OAKS: Arrangements are now being made with a manufacturer for its production in large quantities.

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[2] Public Service Elec. and Gas Co.

MR. STINARD: You spoke of frying and boiling and so on. What is the relative time required for cooking? Apparently your surface temperature would be lower than the electric or gas range.

MR. OAKS: Even though we can produce temperatures of the cooking surfaces up to 700 F, we can find no cooking operation that requires more than 550 F. Incidentally, the highest temperature required in this apparatus is for toasting. Otherwise, the apparatus can be run at an outgoing temperature of 600 F and accomplish all of the services in a successful manner. The cooking time seems to be materially less than either gas or electricity. Just how much, I am not in a position to say, but I should say that it was probably 25 or 30 per cent less than a comparative job done by other means.

1285

No. 1285

A SINGLE-EQUATION DESIGN PROCEDURE FOR RADIANT PANEL SYSTEMS †

By F. W. HUTCHINSON,* LAFAYETTE, IND.

THE RATIONAL design of a panel heating or cooling system involves consideration of many complicating factors which are neglected in the conventional design procedures for non-radiant systems. If adequate engineering accuracy can be obtained in a radiator, convector, or warm air installation which is designed without consideration of such exact rational relationships, one would reasonably expect that comparable accuracy could be obtained from a radiant panel design procedure which was based on the same assumptions. The purpose of the present study is briefly to review the complex rational design procedure which has been developed in a previous A.S.H.V.E. paper [1] and to show, in a step-by-step development, the simplifications realizable from the rational equations if the assumptions underlying conventional non-radiant systems are applied to panel heating and panel cooling.

As will subsequently appear, the rational panel design procedure simplifies from a complex problem requiring simultaneous solution of a minimum of seven equations to the relatively simple task of substituting numerical design data into a single algebraic equation. The single equation is not linear and is somewhat unwieldy in form, but its solution involves no more than a routine substitution followed by arithmetical computation. Thus the problem of panel design is reduced to one of arithmetic, which makes demands on the time, but not on either the skill or judgment of the engineer.

In two respects, the single-equation solution possesses a simplicity and a generality greater than is found in the method of conventional design: (1) Both transmission and ventilation loads are included in the one equation. (In contrast to the procedure for convection design in which these two losses

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† This paper was prepared in cooperation with the Housing Research Division of the Purdue Research Foundation: G. Stanley Meikle, Research Director of the Foundation, and Carl F. Boester, Housing Executive.

* Professor of Mechanical Engineering, Purdue University, Lafayette, Ind. Member of A.S.H.V.E.

[1] Panel Heating and Cooling Analysis by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 285).

Presented at the 52nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1946.

are separately computed.) (2) The comfort relationship is *built into* the equation so that the resultant design is in terms not of an arbitrarily selected inside air temperature, but rather of that particular *comfort air temperature* which is a characteristic of the particular heat transfer relationship of the given structure.

The principal disadvantage of the single-equation procedure is that numerical computations are tedious even though straightforward. The number and inter-relationships of the dependent variables are such that the equation involves addition and subtraction terms which necessitate at least three place accuracy. For the designer who works regularly with radiant panel problems, this disadvantage can be readily overcome by using the equation to establish generalized graphical solutions based on the outside design air temperature for the locality in question and for the fixed maximum permissible panel surface temperature (determined by the structural characteristics of the panel which is to be used). Brief consideration will demonstrate that a complete set of design graphs, based on the equation, could be established such that the problem of panel heating or cooling design could be readily carried out for any locality and for any type or location of panel; other than a determination of the equivalent overall coefficient of heat transfer for the room in question, no computations or reference to equations would then be necessary.

The body of this paper is divided into three parts: I. Critical review of the complex rational method of panel analysis with specific reference to the factors which are to be neglected. II. Development of the equivalent conductance concept and resultant reduction of the rational equations to simpler form and to lesser number. III. Simultaneous solution of the three basic equations eliminating the inside air temperature and inside unheated surface temperature as variables and leading to a single equation expressing the required panel area as a function of panel temperature and outside air temperature (known design constants) and the conductance and ventilation rate (constants of the structure).

I. ANALYTICAL PROCEDURE FOR RATIONAL PANEL DESIGN

The rational procedure for design of a panel heating or cooling system consists in setting up a series of simultaneous heat balance equations; one equation for a heat balance on an average occupant (the *comfort* equation), one for a heat balance on the ventilation air passing through the room and n equations giving heat balances on the n unheated room surfaces which differ in construction, exposure, or orientation (with respect to the vertical). The heat balance on any infinitesimal area of inside unheated surface can be written in general and exact form as,

$$\text{Radiation} + \text{Convection} = \text{Conduction}$$

$$\sigma dA_m \sum_{n=1}^{n=\infty} K_{nm}\left[\left(\frac{T_n}{100}\right)^4 - \left(\frac{T_m}{100}\right)^4\right] + h_m dA_m(t_a - t_m = C_m dA_m(t_m - t_a) \quad . \quad . \quad (1)$$

where subscript $_m$ identifies the infinitesimal area of inside surface on which the heat balance is established while subscript $_n$ (having an infinite number of

point values) identifies each infinitesimal area of the enclosure other than the receiving surface. Further,

σ = Stefan-Boltzmann constant = 0.173
dA = an infinitesimal area
T = surface temperature in deg F absolute
t_a = inside air temperature, deg F
t_o = outside air temperature, deg F
t_m = temperature, deg F of, surface on which the heat balance is established
h = film coefficient of heat transfer, Btu/(hr) (sq ft) (deg F); by convection only
C = conductance of wall from inside surface to outside air, Btu/(hr) (sq ft) (deg F)
K_{nm} = the fraction of radiant energy leaving dA_m which is received by dA_n; by the reciprocity theorem, this is also equal to $\frac{dA_n}{dA_m}$ times the fraction of radiant energy absorbed at dA_m of that leaving dA_n

In order to reduce Equation 1 to a form suitable for use in solving practical problems, means must be found in reducing the infinite series which constitutes the first, or radiation, term. A first step in this direction is to increase dA_m to the largest permissible finite value ΔA_m. Examination of the right side of Equation 1 shows that the conductance limitation on the receiving area is not serious since ΔA_m can be enlarged to a value equal to that of the area of any surface over which the conductance does not vary. Similarly, the convection limitation on ΔA_m can be determined by examination of the second term of the equation. From this term, it is evident that the receiving surface is limited to an area over which the inside film coefficient of convective heat transfer, h, remains constant. Actually, h varies with the texture of the surface, the absolute surface temperature, the air-to-surface temperature difference, the air velocity, the proximity of other surfaces, and the aspect of the surface; in practice, however, the last factor—position of surface with respect to the air—is the only one that appreciably influences the film coefficient for the range of operating conditions that occur in comfort heating or cooling installations. As a first approximation, the film coefficient can be taken as 0.4 or 0.7 or 1.1, respectively, for a horizontal surface facing down (ceiling), a vertical surface, or a horizontal surface facing up (floor). Thus, combining the limitations on surface areas as imposed by conductance and convection, the maximum permissible value of ΔA_m would appear to be the largest section of homogeneously constructed and exposed structure which is found in ceiling, or wall, or floor. An ordinary room would, therefore, divide into six sections of surface consisting of exterior wall, unheated ceiling, inside partition, window area, floor, and panel; these areas will henceforth be designated by the subscripts $_{w,\ c,\ i,\ g,\ f,\ p}$, respectively.

The foregoing discussion leading to a reduction in the number of values of $_n$ in Equation 1 from ∞ to 5 implicitly assumes constancy of temperature over each of the six basic surfaces of the enclosure. This assumption is incorrect for three reasons:

1. In the usual case, there will be a floor-to-ceiling air temperature gradient hence a floor-to-ceiling variation in surface temperature of walls and partitions. This variation is not taken into account in convective design procedure except through selection of a mean design value of the inside air temperature; since the gradient in a

panel installation is usually less than for a conventional system, the justification for neglecting this variation in panel design seems evident.

2. Variation in point-to-point radiant exchange between the receiving area and the various other *unheated* surfaces of the enclosure. This factor appears, for example, in the local cooling effect experienced on a section of unheated ceiling which is adjacent to a wall in which there is a large glass area. The justification for neglect of this factor is not evident, but since it is neglected in conventional design (where its influence is equally as great as for radiant design) a similar assumption of unimportance will be made here.

3. Variation in point-to-point exchange between the panel and the receiving area. This factor may be of considerable importance in establishing marked temperature variation over a surface near the panel (as the upper third of a wall or partition adjacent to a ceiling panel) but it does not have controlling influence on surface temperature variations of average enclosure surfaces. In a room 20 ft x 20 ft x 10 ft, for example, the variation in direct irradiation received at points along an axis of the room from a heating panel occupying the entire ceiling would be approximately 30 per cent, but the maximum fraction of radiant energy leaving the ceiling and incident on a one square foot area centered on one room axis would be slightly less than 15/100 of one percent in general, this degree of irradiation would be small with respect to the convection and conduction transfers occurring from the same unit area.

In addition to the reasons given, neglect of temperature variation over each of the six basic surfaces can be further justified in that the assumed uniform temperature appearing in Equation 1 is a weighted mean value which reflects the influence of radiant exchange as given in the first term of Equation 1. Thus the calculated surface temperatures as used in panel design are not the same as those which would be found from a conventional analysis, but do take account of radiant exchange between the receiver and both the heated and unheated surroundings.

The remaining problem is to investigate the effect of the replacement of dA_m by ΔA_m on the radiation coefficient K_{n1m}. As previously defined, K_{n1m} represents a fraction of energy transfer based on the radiosity (emission plus reflection for an opaque surface) of the source and the actual absorption (rather than irradiation) at the receiving surface. But the radiosity of any area ΔA_{n1} depends on its radiant exchange with each of the other surfaces of the enclosure. The transfer of radiation from ΔA_{n1} to ΔA_m can occur directly, or after 1 to ∞ re-reflections between ΔA_{n1} and ΔA_m, or after 1 to ∞ inter-reflections among ΔA_n^1, the n surrounding surfaces, and ΔA_m; this latter fraction of energy will be received over an infinite number of paths and hence could be evaluated only if the geometry of the system were sufficiently simple to permit establishing boundary conditions for a series of integral equations. Thus, evaluation of K_{n1m} is not simple.

To assist in visualizing the factors which determine K_{n1m}, note that the fraction of energy leaving ΔA_{n1} in the direction of ΔA_m depends only on the geometrical arrangement of these two surfaces with respect to one another. The energy received at ΔA_m either direct from ΔA_{n1} or after re-reflection between ΔA_m and ΔA_{n1} depends on both the geometrical arrangement and the emissivities of these two elements of area. The energy received at ΔA_m after inter-reflection among the n-1 secondary surfaces depends on the factors given plus the geometrical arrangements and emissivities of all n-1 secondary elements of surface.

Fortunately, for the solution of practical problems, the emissivities of surfacing materials other than polished metal are so high (usually in excess

of 90 per cent of black body) that less than 1 per cent of the initially emitted energy goes beyond a second reflection. Because of this rapid absorption of reflected energy, the assumption is commonly made that any enclosure which does not contain polished metal wall surfaces can be treated as though it were made up of non-reflecting materials. The effect of this assumption is to neglect the fraction of energy which reaches ΔA_m after re-reflection or inter-reflection, but to increase the energy reaching ΔA_m by direct transmission. Thus, two compensating departures from actual conditions are involved and the exactness of the approximation will obviously depend on the extent to which, for a particular installation, these two errors offset one another.

When the assumption previously discussed can be justified, the emissivities of the actual surfaces are no longer a factor in establishing K_{nm} and, more important, the geometrical arrangements of the n-1 secondary surfaces do not enter into the evaluation of this constant. The only remaining factor influencing K_{nm} and hence limiting the size of ΔA_m and ΔA_n is the geometrical arrangement of these two surfaces. For this special and simplified condition, the coefficient K_{nm} is said to reduce to the shape factor, F_{nm}, of surface ΔA_n with respect to energy exchanged with ΔA_m; this factor, based on the area of the surface m, is equal to the fraction of energy striking ΔA_n of that emitted by ΔA_m and can be shown [2] to be,

$$F_{nm} = \frac{1}{\Delta A_m} \int_{A_m} \int_{A_n} \frac{\cos \theta_n \cos \theta_m dA_m dA_n}{\pi r^2_{nm}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where:

r_{nm} = distance from dA_m to dA_n
θ_n = angle between r_{nm} and the normal to dA_n
θ_m = angle between r_{nm} and the normal to dA_m

When the source and receiver areas are small the shape factor can be readily evaluated,[3] but for larger values of A_m and A_n the integrations often become difficult; for such cases the numerical values of shape factors can best be obtained from tabulations (for the more common surface arrangements) as given in standard references.[4, 5]

Even with the assumption of an enclosure consisting of black body surfaces, the surface temperature will necessarily vary as a point function since F_{nm} is a point function; however, the average value of F_{nm} for a surface of finite area (as given in Equation 2) can be taken as constant over areas of moderate size and when so used in the heat balance equations will lead to values of the equivalent uniform temperature of each finite surface. For average design conditions in a panel heated or cooled room this series of equivalent temperatures will be adequate for use in computing the mean radiant temperature and hence in analyzing the performance characteristics of the system.

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[2] Loc. Cit. See Note 1.
[3] Simplifying the Calculations for Exchange of Radiant Heat, by B. F. Raber and F. W. Hutchinson. (*Heating, Piping & Air Conditioning*, September, 1944, p. 530.)
[4] *Scientific Basis of Illuminating Engineering*, by Moon (McGraw-Hill, 1936).
[5] *Heat Transmission*, by McAdams (McGraw-Hill, 1942).

Before re-writing Equation 1 in terms of the six basic finite areas, it is helpful to define an equivalent first power coefficient of radiant transfer, h_r, which will permit writing the radiation term without the fourth power exponent. The radiation coefficient is an exact term defined by the equation,

$$h_r = \sigma\left[\left(\frac{T_n}{100}\right)^4 - \left(\frac{T_m}{100}\right)^4\right] \div (t_n - t_m) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

For the range of temperatures customary in comfort installations, h_r is practically constant and has a numerical value of approximately 1.08.

Now re-writing Equation 1 with the simplifying substitutions and in a form to give the heat balance on the exterior wall surface of a room,

$$1.08[F_{iw}(t_i - t_w) + F_{pw}(t_p - t_w) + F_{cw}(t_c - t_w) + F_{gw}(t_g - t_w) + F_{tw}(t_t - t_w) + h_w(t_o - t_w)] = C_w(t_w - t_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Equation 4 constitutes the basic form of surface heat balance equation as used in the previously proposed rational procedure for panel heating or cooling design. As already explained, similar equations must be set up for each different type of non-panel surface. Since in the average room there are five types of such surface and since the inside air temperature and the panel area (for an arbitrarily selected panel temperature) are additional unknowns, it follows that seven independent simultaneous equations must be written and solved. Any further simplification which can be realized from the rational procedure must therefore stem from an improvement in the form of Equation 4; this can most effectively be accomplished by use of the equivalent conductance concept as discussed in Part II.

II. THE CONCEPT OF EQUIVALENT CONDUCTANCE

Transmission losses from a convection heated room are usually calculated by an equation of the form,

$$q = \sum_{n=1}^{n=n} U_n A_n (t_a - t_o) \quad . \quad (5)$$

This equation can be re-written in the form

$$q = \left(\sum_{n=1}^{n=n} U_n A_n \div \sum_{n=1}^{n=n} A_n\right) \sum_{n=1}^{n=n} A_n (t_a - t_o) = U_e A_e (t_a - t_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where

$A_e = \sum_{n=1}^{n=n} A_n$ = total unheated surface of the enclosure (inclusive of such non-transmitting surfaces as inside partitions but exclusive—in radiant systems—of the heating or cooling panel) and,

$U_e = \sum_{n=1}^{n=n} U_n A_n \div \sum_{n=1}^{n=n} A_n$ = the equivalent over-all coefficient of heat

transfer of a homogeneous enclosure for which the total transmission losses would be identical with those of the actual enclosure. For an average room containing the five basic transmitting areas, the equivalent coefficient would be

$$U_e = \frac{U_f A_f + U_c A_c + U_w A_w + U_g A_g + U_i A_i}{A_f + A_c + A_w + A_g + A_i} \quad \ldots\ldots\ldots\ldots \quad (7)$$

while for enclosures in which the temperature on the exterior of the transmitting surface has different values (as for attics, basements, or partially heated adjoining rooms) the equation would take the general form,

$$U_e = \sum_{n=1}^{n=n} U_n A_n (t_a - t_{o_n}) / A_e (t_a - t_o) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (8)$$

where t_{o_n} is the air temperature exterior to transmitting surface n.

The concept of an equivalent over-all coefficient does not reduce the work required for a non-radiant design, but it is of value in that it provides a single index which reflects the heat transfer characteristics of the structure independently of its size or exposure. Thus a comparison of the U_e terms for any two houses will show immediately their relative thermal effectiveness.

When used in radiant design, however, the concept of an equivalent homogeneous enclosure leads to very great simplification. In such an idealized enclosure the surrounding surfaces consist of but two parts: (1) the panel; (2) the equivalent homogeneous unheated room surface. Thus, only one surface heat balance equation need be written and the number of simultaneous equations required for a solution is reduced from seven to three. Further, the panel in the equivalent room will necessarily be identical in size and temperature with the panel required for the actual room, but radiant exchange from the equivalent panel can be expressed in terms of the simple geometrical arrangement of a small surface completely enclosed by a large surface. For this case, the shape factor based on panel area is unity and based on area of the equivalent surround is equal to the ratio A_p/A_e. The equivalent homogeneous room visualization, therefore, serves both to reduce by more than one-half the number of equations needed, and to completely eliminate the task of evaluating shape factors.

A simplified form of Equation 4 can now be written as a heat balance on the homogeneous unheated (or cooled) surround,

$$1.08(A_p/A_e)(t_p - t_a) + h_e(t_a - t_o) = C_e(t_a - t_o) \quad \ldots\ldots\ldots\ldots \quad (9)$$

where C_e is the equivalent conductance of the enclosure and can be calculated from the equivalent over-all coefficient by the equation

$$C_e = \frac{1}{\frac{1}{U_e} - \frac{1}{1.65}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

The combined radiation-convection film coefficient (1.65) is used in Equation 10 because of the fact that this is the value used in computing the over-all

coefficients which are listed in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1945. The film coefficient for convection only, h_c, will obviously have some average value dependent on the relative amount of enclosure surface which exists in wall, floor, and ceiling; since the range of this coefficient is from 0.4 to 0.8 to 1.1, a reasonable average value suitable for general use is 0.7 Btu/(hr) (sq ft) (deg F).

Substituting the numerical value of h_c and letting u and v represent the fractions, respectively, of total enclosure surface which are panel or non-panel, Equation 9 becomes,

$$1.08u(t_p - t_a) + 0.7v(t_s - t_a) = C_e v(t_e - t_o) \quad \ldots\ldots\ldots\ldots (11)$$

The second of the three required simultaneous equations can be obtained by writing a heat balance on the ventilation air passing through the room,

Convection gain from panel − Convection to unheated surface
= Energy gain of ventilation air

$$h_p A_p(t_p - t_a) - 0.7A_e(t_s - t_a) = 0.24W(t_a - t_o) \quad \ldots\ldots\ldots (12)$$

where:

h_p = film coefficient of convective heat transfer from panel to room air
A_p = panel area
0.24 = specific heat of air at constant pressure
W = pounds per hour of outside ventilation air passing through the enclosure.

Re-writing Equation 12 in terms of u and v,

$$h_p u(t_p - t_a) + 0.7v(t_s - t_a) = 0.24 \frac{W}{A_R + A_e}(t_a - t_o) \quad \ldots\ldots\ldots (13)$$

Then letting $V_e = \dfrac{13.5W}{A_p + A_e}$ = cubic feet of outside ventilation air passing each square foot of inside surface per hour, and making this substitution in Equation 13,

$$h_p u(t_p - t_a) + 0.7v(t_s - t_a) = 0.018V_e(t_a - t_o) \quad \ldots\ldots\ldots\ldots (14)$$

The third independent equation is an empirical one[6] which, in effect, sets up a heat balance on the occupant. In its most generally accepted form, this *comfort* equation is linear and can be written,

$$t_a = a - b(\text{MRT}) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (15)$$

where a and b are design constants which usually are assigned values of 140 and unity, respectively. The mean radiant temperature is a complex term,[6]

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[6] Radiant Energy Exchange as a Factor in Airplane Cabin Heating, by B. F. Raber and F. W. Hutchinson (*Journal of the Aeronautical Sciences*, July, 1944).

but is frequently approximated as an average surface temperature,

$$\text{MRT} = ut_p + vt_e \quad \ldots \ldots \ldots

where

$$K_1 = a - but_p \text{ and } K_2 = 1 + bv \quad \ldots \quad (19)$$

or

$$t_o = \frac{K_3 + vt_oC_e}{K_4 + vC_e} \quad \ldots \quad (18c)$$

where

$$K_3 = 0.7vK_1 + 1.08ut_p \text{ and } K_4 = 0.7vK_2 + 1.08u \quad \ldots \quad (20)$$

Equating 14*a* and 17 and solving for t_o,

$$t_o = \frac{(a - but_p)(0.7v + h_pu + 0.018V_o) - (h_put_p + 0.018V_ot_o)}{0.7v + bv(0.7v + h_pu + 0.018V_o)} \quad \ldots \quad (21)$$

$$= \frac{v(a - but_p)0.7 + (a - but_p)(h_pu + 0.018V_o) - (h_put_p + 0.018V_ot_o)}{v(1 + bv)0.7 + bv(h_pu + 0.018V_o)} \quad (21a)$$

$$= \frac{0.7vK_1 + K_1(h_pu + 0.018V_o) - (h_put_p + 0.018V_ot_o)}{0.7vK_2 + bv(h_pu + 0.018V_o)} \quad \ldots \quad (21b)$$

$$= \frac{K_5 + 0.018V_o(K_1 - t_o)}{K_6 + 0.018V_obv} \quad \ldots \quad (21c)$$

where

$$K_5 = 0.7vK_1 + h_pu(K_1 - t_p) \quad \ldots \quad (22)$$

and

$$K_6 = 0.7vK_2 + h_pubv$$

Equating 18*c* and 21*c* and cross-multiplying,

$$K_3K_6 + 0.018V_obvK_3 + vt_oK_6C_e + 0.018V_obv^2t_oC_e = K_4K_5 + 0.018V_o(K_1 - t_o)K_4 + vK_5 + 0.018V_o(K_1 - t_o)vC_e \quad \ldots \quad (23)$$

Solving for C_e,

$$C_e = \frac{[(K_1 - t_o)K_4 - bvK_3]0.018V_o + (K_4K_5 - K_3K_6)}{[bv^2t_o - v(K_1 - t_o)]0.018V_o + (vt_oK_6 - vK_5)} \quad \ldots \quad (24)$$

$$C_e = \frac{0.018V_o(K_1K_4 - t_oK_4 - bvK_3) + (K_4K_5 - K_3K_6)}{v[0.018V_o(bvt_o - K_1 + t_o) + (t_oK_6 - K_5)]} \quad \ldots \quad (25)$$

where (as from Equations 19, 20, 22)

$K_1 = a - but_p$ $\quad K_3 = 0.7vK_1 + 1.08ut_p$ $\quad K_5 = 0.7vK_1 + h_pu(K_1 - t_p)$
$K_2 = 1 + bv$ $\quad K_4 = 0.7vK_2 + 1.08u$ $\quad = K_1(0.7v + h_pu) - h_put_p$
and $u + v = 1$ $\quad K_6 = 0.7vK_2 + h_pubv$

Equation 25 gives a panel design solution in a form equivalent to,

$$u = \theta(t_p, t_o, V_c, C_e, a, b, h_p) \quad \ldots \quad (26)$$

where

t_p = a design constant equal to the maximum permissible panel surface temperature which will occur at a time when the load on the system is greatest.
t_o = design (minimum outside air temperature corresponding to a condition of maximum load on the system.

and V_c, C_e are structural design constants fixed by thermal characteristics and ventilation (or infiltration) characteristics of the enclosure, while b and a are

empirical coefficients of the comfort equation (usually taken as 1 and 140 respectively, and h_p is the film coefficient of *convective* heat transfer from the panel and is constant at 0.4, 0.7, or 1.1 depending on whether the panel is located in ceiling, wall, or floor, and K_1, K_2, K_3, K_4, K_5, K_6, are arbitrarily defined parameters selected in order to reduce the procedure of computation to a direct and orderly basis.

For a given locality and a selected type of panel, Equation 26 reduces to the form,

$$u = \theta^1(V_c, C_e) \quad \ldots \quad (27)$$

Equation 27 is obviously amenable to simple graphical solution and could readily be used as a basis for preparing sets of graphs which would reduce the problem of panel design to a routine matter of following a dashed line to the answer.

One advantage possessed by Equation 25 in the form in which it is written is that it provides a straightforward means of determining the boundary insulation requirements for an enclosure in a given locality which is to be heated (or cooled) by means of a particular type of panel. As an example, determine the maximum permissible equivalent conductance of a room 20 ft x 20 ft x 10 ft if it is to be heated by an all-over ceiling panel with maximum surface temperature of 100 F at a time when outside air temperature is 0 F; infiltration is taken as 1 air change per hour:

$$A_p = 20 \times 20 = 400$$
$$A_p + A_e = 2 \times 20 \times 20 + 4 \times 20 \times 10 = 1600$$
$$u = \frac{A_p}{A_p + A_e} = 0.25$$
$$v = 1 - u = 0.75$$
$$V_c = \frac{(20 \times 20 \times 10)1}{1600} = 2.5$$
$$t_p = 100$$
$$t_o = 0$$
$$a = 140$$
$$b = 1$$
$$h_p = 0.4$$

Then

$$K_1 = 140 - 0.25 \times 100 = 115$$
$$K_2 = 1 + 0.75 = 1.75$$
$$K_3 = 0.7 \times 0.75 \times 115 + 1.08 \times 0.25 \times 100 = 60.37 + 27.00 = 87.37$$
$$K_4 = 0.7 \times 0.75 \times 1.75 + 1.08 \times 0.25 = 0.919 + 0.270 = 1.189$$
$$K_5 = 0.7 \times 0.75 \times 115 + 0.4 \times 0.25(115 - 100) = 60.37 + 1.50 = 61.87$$
$$K_6 = 0.7 \times 0.75 \times 1.75 + 0.4 \times 0.25 \times 0.75 = 0.919 + 0.075 = 0.994$$

and substituting into Equation 25,

$$C_{e(max)} = \frac{0.018 \times 2.5(115 \times 1.19 - 0 - 1 \times 0.75 \times 87.37) + (1.19 \times 61.87 - 87.37 \times 0.994)}{0.75[0.018 \times 2.5(0 - 115 + 0) + (0 - 61.87)]}$$

$$= \frac{0.045(71.32) - 13.22}{0.75(-67.04)} = \frac{10.01}{50.28} = 0.199;$$

Then,

$$U_e = \frac{1}{\frac{1}{C_e} + \frac{1}{1.65}} = \frac{1}{5.02 + 0.606} = 0.178$$

Since most rooms have a substantial fraction of total surface in the form of inside partition (with a consequent lowering of the equivalent over-all coefficient) it is evident that the calculated maximum value of U_e for this case is sufficiently great to indicate the feasibility of ceiling panels for rooms meeting the conditions as given in the example.

SUMMARY

The assumptions and approximations used in setting up the simultaneous heat balance equations for the rational procedure of panel heating or cooling design have been considered critically and in detail. Based on these assumptions, a typical surface heat balance equation (4) has been written and the method indicated for establishing the six additional equations needed for an average design problem. The concept of equivalent conductance was introduced and its use as a means of reducing the number of required simultaneous equations from 7 to 3 then demonstrated. Finally, the three equations for an equivalent homogeneous room were written and used simultaneously giving a single-equation (25) design procedure for determining by direct substitution the required size of a given type of panel for any particular enclosure.

Emphasis is placed on the fact that the single-equation solution is simple and direct even though the appearance of the equation is formidable and its numerical solution tedious. Once the design constants have been selected, solution of the equation involves nothing more than routine substitution and progressive numerical condensation of the terms; this is a task which can readily be performed by unskilled personnel. Whenever the frequency of design problems is sufficient to justify an initial expenditure of time equal to that required for the solution of fifteen problems, it will be found economical as well as convenient to set up a graphical solution (for a given locality and given type of panel) of the basic equation.

ACKNOWLEDGMENT

The author wishes to acknowledge with thanks his continued indebtedness for the advice, assistance and helpful criticism of Prof. B. F. Raber.

DISCUSSION

R. G. VANDERWEIL, Waterbury, Conn. (WRITTEN): I have read with great interest Professor Hutchinson's single equation design procedure for radiant panel systems. Unfortunately, I was not in a position to study the method sufficiently to give a complete discussion, but I have tried to familiarize myself enough with these equations to discuss the possibilities of their practical use.

At first, I would like to congratulate Professor Hutchinson on presenting a method which reduces the seven equations of the original method to one. If we examine the computation systems dealing with the room heat flow, we can distinguish between two kinds of methods: One, for example, is the rational method as presented by Professors Raber and Hutchinson and several other authors, and the other is the approximate method. Due to various assumptions, the single equation method falls within the latter group.

It is worthwhile to remember that we have available another simpler method, originally presented in the United States by Dr. F. E. Giesecke, College Station, Tex. It was based on assumptions similar to those made for the *single equation procedure.* Both Professors Giesecke and Hutchinson, as well as others, have decided to replace a room having a number of enclosures, such as ceiling, floor, glass and wall, by a hypothetical room having only two enclosures, namely the heating surface and the heated surface. For obvious reasons, it is much simpler to compute the heat inter-

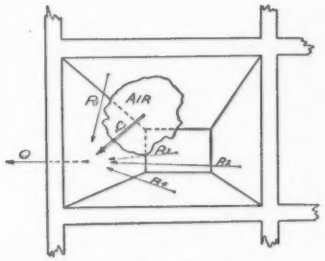

FIG. A. HEAT FLOW BALANCE FOR A SINGLE OUTSIDE WALL

change between two surfaces (and air) than to take into consideration the mutual heat flow from six or more surfaces (and air).

Let us now visualize the heat flow as computed in rational and in simplified methods. Fig. A shows the heat flow balance for a single wall—the outside wall of the actual room only. In the rational method all individual flow components are computed as R_1 to R_4 and C_1—into the wall and the wall's heat loss to the outside, Q_1, is considered equal to the combined heat supplied by the individual components. According to this method it is necessary to establish several other equations, the solution of which yields the room heat loss and all individual wall surface temperatures as well as the air temperature. All of the room's temperatures are computed and with these temperatures known, the individual heat transfer coefficients, h and h_r, are unequivocally determined. For instance, the convective coefficient in a 65 F atmosphere of a 95 F floor is $h = 1.2$; of a 70 F wall is $h = 0.57$; and of a 45 F glass area is $h = 0.8$. The radiation coefficient from floor to the 70 F wall is $h_r = 0.99$; and from the wall to the glass, $h_r = 0.87$, both for a 90 per cent emissivity. By trial and error, all rational methods may be solved to obtain the proper temperatures as well as in the proper conductances h and h_r.

Fig. B. shows the assumptions for all simplified methods. All unheated room surfaces are now combined into one area of *mean* heat flow properties A_u, which is

opposed by the heated area, A_h. This assumption results in very simple heat flow conditions. All components of heat flow in such a room are represented by only three arrows.

Let us now disregard all rational methods and restrict our discussion to the possible assumptions when using simplified methods as given by Fig. B.

(*a*) Professor Hutchinson arrives at temperatures by assuming for all but the heated surface a constant convective film coefficient of 0.7 Btu per (hr) (sq ft) (deg F) and a radiant conductance of 1.08 Btu per (hr) (sq ft) (deg F). The first coefficient may change as much as 250 per cent (from 0.4 to 1.1) depending upon location of the surface considered and upon the temperatures involved. The latter as much as 70 per cent (from 0.7 to 1.2). For obvious reasons, any method assuming mean values of variables changing to such a degree, cannot give entirely correct results and if we derive temperatures and heat flow according to said assumptions, we must more or

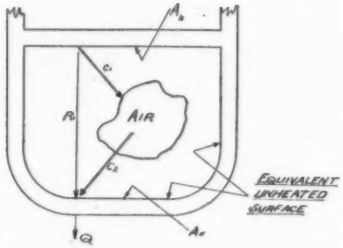

FIG. B. ASSUMPTIONS USED IN SIMPLIFIED METHODS FOR RADIANT HEATING COMPUTATIONS

less deviate from the actual conditions according to the deviation of the assumed conductances from the actual conductances. Thus it seems as though errors in the magnitude of 15 to 20 per cent may possibly be met with by using this method.
(*b*) In the second group of simplified methods, such as Professor Giesecke's, one suggested by myself, and others, the temperatures are assumed and the conductance derived. Professor Hutchinson assumes the conductance and derives the temperatures. For obvious reasons, when assuming the temperatures rather than the conductances, the errors must be greater—possibly as great as 25 to 30 per cent. However, by assuming the temperatures properly we may be sure to err always in the conservative direction and then we can afford to make errors.

To summarize, we must state that the single equation method is definitely superior. However, time is another factor to be considered. When computing radiant heating systems, we are confronted with the room heat flow problem as well as with the problem of design of the coil (tube diameter, tube centers and water temperature) and the piping system. All of these steps take considerable time. For this reason, I believe the designer will give serious thought to using the less correct method which gives results in about 40 sec rather than the superior method which will require about 40 min computing time per room. From the viewpoint of the consulting engineer or contractor who is rushed to turn out one layout after another, I must say that an even simpler computation method is desirable and I, personally, hope that Professor Hutchinson will, at a later date, present us with such a method.

Concerning the use of the single equation method, I wonder if it would not be advisable to use values other than *means* (0.7 and 1.08) for convective and radiant conductances. In order to obtain conservative results under all circumstances, it may be of advantage to take more conservative values for these two coefficients.

L. P. SAUNDERS: I think it is most interesting and probably very gratifying to Professor Hutchinson to have so much interest shown. Invariably when a purely technical subject is discussed the meeting is not very well attended. Am I to assume that Professor Hutchinson has silenced everyone?

MR. VANDERWEIL: I would like to ask whether Professor Hutchinson's single equation method agrees so well with the seven equation method because the latter is also based on mean rather than on actual radiative and convective conductances, that is, h and h_r of the paper?

PROFESSOR HUTCHINSON: The coefficient h_r is an equivalent coefficient which strictly speaking should never be used. The term h_r is an equivalent term, based on the Stefan-Boltzmann four-power law. It involves also the visibility factor, the shape factor of U or of the object in question, with respect to the panel and with respect to every other surface of the room. It also involves variations in the surface emissivity of each of the unheated surfaces throughout the entire room. Consequently, the so-called equivalent radiation factor, h_r, which I believe is taken here as 1.08, is basically a term which in itself has no direct significance. It is a functional term of exactly the same type as U. You cannot say, strictly speaking, that h is something which physically you can see. You can say h is a number which represents the combined influence of about seven different terms. It is entirely correct that the agreement we found is based upon the assumptions that the coefficients were the same in both cases. Obviously, however, we wouldn't change the coefficients since we collected them in the first place based upon a study of the h_r shape factor analysis and so on, to the best of our judgment.

I believe that, on the basis of a six months study, 1.08 was recommended in the original paper (1941). Since then I have not found any evidence which would lead me—and I believe I can also speak for Professor Raber—to change my mind. Consequently it is very true that the agreement is based upon the fact that we have not changed our minds.

W. L. FLEISHER, New York, N. Y.: I think the Society is prone to believe that panel heating is the only method of heating. I feel that we are emphasizing to a great extent not only panel heating or radiant heating but an exclusive method of performing that particular function. For instance, we find in *Life Magazine* and in many other magazines, information giving the impression that pipes running through floors and ceilings will form the only system by which our future homes will be heated. Of course, everybody knows that this is the oldest method there is, and that in the beginning this type of heating was done without any kind of pipe, to give perfectly good results.

Now, I assume that Professor Hutchinson will tell you that the distinction between radiant heating and panel heating is simply a difference in the method of heating the radiating surface. He might also add that any method by which the surfaces which surround people are heated to a temperature above outside conditions will perform a function which may be as good as those that have been developed by various interested persons, and by himself.

So I simply express a warning that we are possibly carried away by our enthusiasm and are, therefore, emphasizing this particular method of heating to a greater extent than it should be emphasized.

You heard Mr. Corbett say yesterday that when we get panel heating we will sit in our shirt-sleeves at a temperature below that at which any of us enjoy sitting in our shirt-sleeves now. He also said that we will not need any air. We should not let ourselves be carried away too much by such talk, instead let us ask Professor

Hutchinson whether any other method of heating his panels will not give equivalent results.

PROFESSOR HUTCHINSON: Mr. Fleisher, I want to call particular attention to one thing which I think possibly you overlooked. First, I am in complete agreement with everything that you have said, but I do want to get out from under one remark and that is the implication that I am in any way interested in or ever have been in any particular method.

I want to call particular attention to the fact that in the work we have done at Berkeley originally and are doing now in cooperation with the Society we have made a strong point of never even mentioning a method. We never go back of the surface.

This paper and the preceding papers have without exception, never gone back of the surface. We have talked only in terms of the area needed and the temperature. Obviously, whether you do it electrically, or by hot air or with pipes, or any other method, makes no particle of difference. It does make a difference to the man who puts it in, but does not affect us in the things that we are doing. I am in no sense of the word an advocate of any type of panel heating system and I do not think I have ever given justification for even being suspected of being an advocate of panel heating. If I were to build a house I do not think it could be taken for granted that I would use panel heating. As a research man I am working on panel heating research in connection with surface temperatures but I hope that will not be held against me or held to constitute evidence of the fact that I either support a particular method or support the system. I do not support either one or the other.

Our interest has been purely in terms of surface. Mr. Fleisher is a thousand per cent correct in everything he says. I missed the session yesterday but you quoted someone as saying that we will be able to sit in air at very low temperature without having any air. I may call attention to a discrepancy of that same kind in terms of heat balance. If you sit in any room with zero air conditions the air temperature is going to equal the mean radiant temperature. It can not possibly do otherwise,—by a simple heat balance on the system. I have yet to see anything in the way of a sufficient air change to provide any reduction in air temperature below 60 F, and it is unusual to get down to 60 F. In my opinion the concept of panel heating leading to air temperatures of 55 F or 50 F is absolute nonsense. It is achievable if you are willing to pay hundreds of dollars per month to maintain your heating system, pour thousands of cubic feet of air through by power, and if you are willing to have the rugs blown off the floor and so on.

Commenting on Mr. Fleisher's remarks I would say very definitely in my opinion there is no possibility in normal panel heating of ever achieving anything like the air temperature reductions that some enthusiasts have said are going to be possible. I do not believe it.

C. S. LEOPOLD, Philadelphia, Pa.: I agree with Mr. Fleisher in his warning concerning over-emphasis of panel heating. In defense of Professor Hutchinson, I recall that in one of his early papers on panel heating he made this statement: *Panel heating differs from radiator heating more by intent than result.* Is that correct, Professor Hutchinson?

MR. FLEISHER: I really should say a word. I got up to state that I in no way wanted to criticise Professor Hutchinson, I certainly enjoy, as much as does anyone else here, his discussion of any of these papers.

MR. VANDERWEIL: I wish to emphasize that in all points but one, I do agree with Professor Hutchinson's paper. The only exception is that for practical purposes I suggest the desirability of an even simpler solution. I came out here to discuss the paper in the hope of suggesting to Professor Hutchinson that a continuation in the direction of simplifying the method would be greatly appreciated by all those involved in radiant heating.

In order to show how much we can simplify the design procedure by using the approximate method in which the temperatures are assumed, let us look at Fig. C. In this figure, the ordinate represents the mean effective room temperature (65 F in the average home, 58 F in average industrial building) and the abscissa, the panel output in Btu per (hr) (sq ft of panel). The sloping lines represent panel surface temperatures.

When using the graph it is necessary only to divide the room heat requirement (say, 10,000 Btu/hr) by the area of the surface to be provided with a heating coil

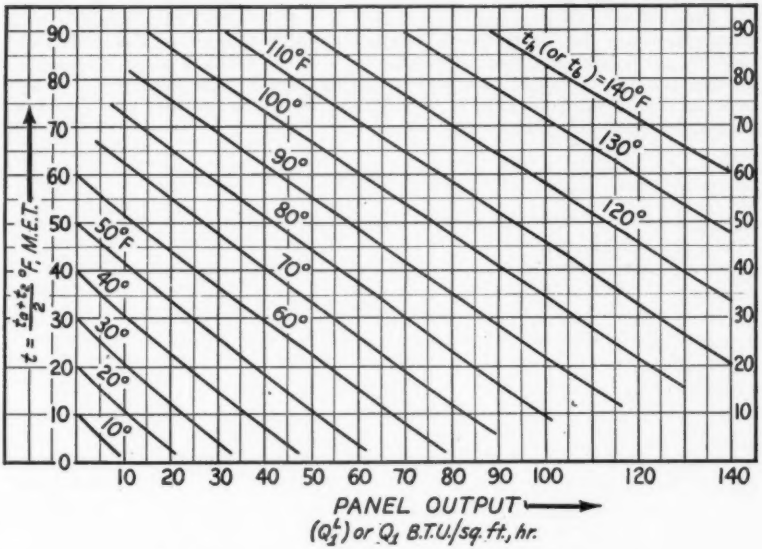

FIG C. CEILING PANEL TEMPERATURE AND OUTPUT

(say 200 sq ft). The required unit heat output is then obtained as 10,000/200 = 50 Btu (hr) (sq ft). From the known mean effective temperature of 65 F for example, you run one horizontal line through $t = 65$ F and one vertical line through $Q = 50$ Btu per sq ft and obtain the desired panel surface temperature of $t_h = 98$ F.

I developed this particular graph to be used on ceiling panels and in rooms having natural ventilation (or mechanical ventilation supplying air of room air temperature) only. I agree with Professor Hutchinson that these graphs present anything but a rational solution. The same is true about a great many data used for heating layouts, but from many a check we know that, although a few panel surfaces proved to be over-dimensioned, the graphs proved reliable for all practical purposes.

Concluding, I am sure that all of us would welcome a more correct method of computation that gives results per room within a few minutes and which may be used with ease on larger projects. But again, I would like to express appreciation to Professor Hutchinson for his having condensed the complicated theory sufficiently to be expressed by a single equation.

PROFESSOR HUTCHINSON: One slight point I had better qualify and say that Mr. Vanderweil and I have talked about this for so many years and disagreed for so many years that he has no hope of persuading me, nor I him.

I call your attention to one thing, however, if we are going to talk about approximate solutions, why not use the approximate solution all of you know? Why not calculate the load in the usual way? Then reach your panel correspondingly. The only reason for not using the usual methods on panel design is that many of you have thought they were not satisfactory. They do give an approximate solution. Since we are looking for something a little better than an approximate solution, I feel that anything which is not better at least than the existing method of just calculating the losses perhaps does not deserve any more consideration than the existing method. Mr. Leopold, Mr. Fleisher, Professor Raber, and many others as a result of their work over many years in the profession, have contributed much on this subject and much of my information comes from studies of their work. I insist, although so far they might deny it, that I am merely quoting them in large measure. It is true in actual panel heating work but it stems back to the things that they have done over such a long period of time.

1286

No. 1286

LOW-PRESSURE PROPERTIES OF WATER FROM −160 TO 212 F

By JOHN A. GOFF * AND S. GRATCH,** PHILADELPHIA, PA.

INTRODUCTION

IN CONNECTION with a cooperative investigation into the thermodynamic properties of moist air, between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Towne Scientific School, University of Pennsylvania, it became necessary to examine critically available data on the properties of water itself. In this country the accepted source of information regarding the thermodynamic properties of water is the Keenan-Keyes steam tables [1] published in 1936, hereinafter referred to as *the steam tables*. Critical examination of the data given in these tables revealed that they do not possess a very high degree of accuracy at temperatures below 212 F; hence, an attempt to improve them in this range became necessary. The present paper outlines this attempt at improvement and offers important revisions to Table 1, Saturation (liquid-vapor), in the range 32 to 212 F, and to Table 5, Saturation (solid-vapor). The range of Table 5 has been extended downward to −160 F.

The present paper is offered as a contribution from the University of Pennsylvania Thermodynamics Research Laboratory recently established at the University of Pennsylvania with the active support of the Navy Department, Bureau of Ships.

SATURATION PRESSURE

In order to achieve accuracy in the formulation of the thermodynamic properties of moist air, as published in a recent paper,[2] it was especially important to have accurate information regarding the saturation pressure p_s and of the second virial coefficient A_{ww} of water as functions of absolute temperature T. The coefficient A_{ww} appears in the equation,

$$pv = RT - A_{ww}p \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (1)$$

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[1] Exponent numerals refer to Bibliography.

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valid at sufficiently low pressures p. At temperatures below about 150 F, this equation appears to be valid from zero pressure to saturation pressure; accordingly, it is accurate to write

$$p_s v_g = RT - A_{ww} p_s \quad \ldots \quad (2)$$

at these temperatures. Using the value of RT_o, namely 22414.6 ± 0.6 (cm^3atm/mole) recommended by Birge [3] (1941), where T_o denotes ice-point temperature, also using what appears to be the best value for the molecular weight of water, namely, 18.0160 (lb/mol), the value,

$$A_{ww} p_s = 24 \pm 42 \text{ (ft)}$$

is readily computed from values of p_s and v_g listed in the steam tables at 32 F. The uncertainty ± 42 (ft) is that produced solely by the tolerance of ± 0.1 per cent assigned to the saturation pressure at 32 F by the Third International Conference on Steam Tables.[4]

At 158 F, the tolerance assigned to the steam table value of saturation pressure is only ± 0.05 per cent. Using the value of T_o recommended by Birge,[3] namely, 273.16 ± 0.01 K, and assuming a linear relation between absolute temperature T and international centigrade temperature t, the value,

$$A_{ww} p_s = 377 \pm 26 \text{ (ft)}$$

is readily computed from values of p_s and v_g listed in the steam tables at 158 F. Again the uncertainty of ± 26 (ft) is that produced solely by the tolerance assigned to the saturation pressure.

The foregoing calculations make it clear that, if the accepted tolerances in the steam table values of saturation pressure were the only source of uncertainty, fairly satisfactory values of second virial coefficient A_{ww} might be obtained from steam table data for the higher temperatures, but not for the lower.

SATURATED-VAPOR VOLUME

Steam table values of the specific volume of the saturated vapor v_g were computed from the empirical equation of state constructed by Keyes, Smith, and Gerry [5] to fit their experimental p,v,t-data in the range 195 to 460 C. The extrapolation to temperatures below 212 F must be expected to introduce considerable uncertainty. In deed, Collins and Keyes [6] have found discrepancies of about 15 per cent at 104 F, 15 per cent at 140 F, 6 per cent at 212 F, between their measured values of the coefficient $(\delta h/\delta p)_T$ and corresponding values computed from the Keyes, Smith, and Gerry equation of state by means of the thermodynamic identity,

$$(\delta h/\delta p)_T = (\delta v_T/\delta T)_p \quad \ldots \quad (3)$$

where τ denotes reciprocal absolute temperature. In view of these discrepancies, uncertainties of at least ±0.04 per cent at 104 F, ±0.08 per cent at 140 F, ±0.10 per cent at 212 F, must be assigned to the steam table values of v_g.

From the previous calculations it seems fair to conclude that the uncertainty in the steam table values of vapor volume v_g is of the order of ±0.10 per cent over most of the range 32 to 212 F. This greatly reduces the reliability of values of A_{ww} computed directly from steam table data.

SATURATED-VAPOR ENTHALPY

The steam table values of saturated-vapor enthalpy h_g were obtained by computing the coefficient $(\delta h/\delta p)_T$ from the Keyes, Smith, and Gerry equation of state by means of the thermodynamic identity Equation (3), integrating this coefficient from zero pressure to saturation pressure p_s, and adding values of zero-pressure enthalpy calculated by Gordon [7] from the spectroscopic data of Mecke [8] and coworkers. But Gordon's values are probably in error by as much as 0.5 Btu per lb in part of the range 32 to 212 F because he neglected to include the effect of centrifugal distortion of the water molecule in his calculations. Newer spectroscopic data and recent revisions in the fundamental physical constants (Planck's constant h, Boltzmann's constant k, Avogadro's number N_o) suggest an even greater probable error. Furthermore, as explained previously (Saturated-Vapor Volume), the values of $(\delta h/\delta p)_T$ derived from the Keyes, Smith, Gerry [5] equation of state are not very reliable, and contribute to a probable error of at least 0.3 Btu per lb at 212 F.

The cumulative probable error in the steam table values of saturated vapor enthalpy h_g can be estimated by comparing the steam table value 1075.8 Btu per lb with the value 1075.1 ± 0.1 Btu per lb given by Osborne, Stimson, and Ginnings [9] for 32 F.

OTHER PROPERTIES

Steam table values of saturated vapor entropy s_g are probably no more reliable than those of saturated vapor enthalpy h_g; those of saturated liquid volume v_f in the range 32 to 212 F are probably more than sufficiently accurate for all practical purposes; those of saturated liquid enthalpy h_f and of saturated liquid entropy s_f are also quite accurate but should be revised in the light of more recent, extremely accurate, Bureau of Standards values.[9]

Values listed in Table 5 of the steam tables are entirely inadequate both as to accuracy and as to range. This table should be reformulated and extended to cover a much wider range of temperature from 32 F downward.

ZERO-PRESSURE PROPERTIES OF WATER VAPOR

Mecke,[8] et al, were the first to fully analyze the rotation-vibration spectrum of the water molecule. Their results have since been extended and improved by the use of theoretically more rigorous analytical methods. The best information presently available is that of Randall, Dennison, Ginsburg, and Weber,[10] of Darling and Dennison,[11] and of Nielsen.[12] These investigators have analyzed higher rotational levels than did Mecke and have taken account

of such effects as centrifugal distortion and certain perturbations between vibrational levels, which effects had been neglected by Mecke.

Gordon [7] calculated the zero-pressure properties of water vapor from the spectroscopic data of Mecke and coworkers [8] using the now obsolete values of the fundamental physical constants given in *International Critical Tables.* Wilson [13] called attention to the fact that Gordon had neglected to take account of the centrifugal distortion of the water molecule and estimated a correction for this effect to be applied to Gordon's values. Stephenson and McMahon [14] calculated this correction for temperatures below 300 K directly from the newer spectroscopic data of Randall, Dennison, Ginsburg, and Weber [10] and found agreement as to form, but slight disagreement as to magnitude, with Wilson's estimate. To obtain accurate values of the zero-pressure properties, especially at the higher temperatures, clearly calls for a systematic recalculation based on newest spectroscopic data and best values of the fundamental physical constants.

Recently the *American Society of Mechanical Engineers* has authorized the formation of a Special Research Committee on Gas Properties which will undertake to assemble information regarding the thermodynamic properties of gases and gas mixtures of sufficient accuracy to warrant acceptance as standard. Undoubtedly this Committee will want to sponsor the systematic recalculations referred to.

The temperature range of interest in the present paper extends only from 100 to 400 K. Throughout this range true equilibrium between the *ortho* and *para* modifications of the water molecule is always attained so that the composition of ordinary water as a mixture of these two modifications is fixed and definite. Also, in this range the rotational levels are fully excited, according to Stephenson and McMahon, and small changes in the vibrational levels from those used by Gordon, as well as certain perturbations neglected by him, do not affect the final results appreciably. Finally, in this range the correction for centrifugal distortion as determined by Stephenson and McMahon is sufficiently accurate.

In the present paper only the variations with temperature, not the absolute values of enthalpy and entropy are of interest because the usual steam-tables practice of assigning each the value *zero* for saturated liquid at 32 F is to be followed. Since the rotational levels are fully excited in the temperature range considered, small changes in the rotational constants from those used by Gordon would alter the calculated values of enthalpy and entropy by constant amounts only and may therefore be ignored.

In view of these circumstances, sufficiently accurate values of zero-pressure enthalpy and reduced entropy can be obtained from the Gordon data by correcting them for changes in the fundamental physical constants and for the effect of centrifugal distortion. The most satisfactory procedure is first to construct a semi-theoretical control equation whose constants can be adjusted to fit the Gordon data, then to revise these constants as required by changes in the fundamental physical constants, finally to add the correction for centrifugal distortion. From the resulting formula the zero-pressure properties of interest can be calculated at as close intervals of temperature as desired.

Analysis of the spectrum of a molecule determines a set of discrete energy levels ϵ_i representing the only energy values the molecule can assume in its various modes of internal motion—rotational, vibrational, electronic, nuclear.

Actually, the wave numbers ν_i not the energy levels themselves, are the quantities directly measured; but these are related by the simple formula, $\epsilon_i = h c \nu_i$, where h denotes Planck's constant and c, the speed of light. The sum over all internal quantum states (more than one distinct quantum state may have the same energy level) of the quantity $e^{-\epsilon_i/kT}$ is called the *internal partition function* and is denoted by Q_{int}.

If the water molecule were a rigid rotator whose atoms vibrate about their equilibrium positions with such small amplitude as to be sensibly harmonic and not to alter the moment of inertia of the molecule appreciably, the complete partition function Q, which includes a factor representing the contribution from translational motion, would be given by the expression,

$$lnQ = -3ln\tau + \sum_{j=1}^{3} ln\left(\frac{e^{\theta_j/\tau}}{e^{\theta_j/\tau} - 1}\right) + b\tau + c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4a)$$

provided the rotational levels are fully excited, as is the case at temperatures above 100 K. The constants b and c are to be regarded as adjustable, but the *characteristics temperatures* θ_j are to be calculated from the measured fundamental vibrational *frequencies* ω_j as follows:

$$\theta_j = (hc/k)\omega_j \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .

Column 1, Table 1, are listed the adjusted values while in Column 2 are listed the values calculated from the spectroscopic data used by Gordon, for comparison.

The zero-pressure *mean* specific heat $(h^\circ - \bar{u})/T$, where $\bar{u}$ denotes null-point energy, is derived from Q by

$$\frac{h^\circ - \bar{u}}{T} = -R\tau \frac{dlnQ}{d\tau} \quad \ldots \quad (5b)$$

and therefore contains b, but not c. Least squares adjustment of b to fit Gordon's values of mean specific heat in the range 298.1 to 1500 K gave a root mean square deviation of 0.0006 and a maximum deviation of 0.0012.

TABLE 1—SPECTROSCOPIC CONSTANTS

COLUMN	1	2	3
θ_1 (°R)	4124.09	4111.2	4141.6
θ_2 (°R)	9317.47	9279.0	9356
θ_3 (°R)	9802.06	9682.2	9844
a_1	−0.03958	−0.0477	−0.03958
a_2	0.05353	0.0321	0.05353
a_3	0.04000	0.0342	0.04000

Finally, the zero-pressure reduced entropy $(s + Rlnp)^\circ$ is derived from Q by

$$(s + Rlnp)^\circ = (h^\circ - \bar{u})/T + RlnQ \quad \ldots \quad (5c)$$

and therefore contains both b and c. With the former already adjusted, least squares adjustment of the latter gave a root mean square deviation of 0.0004 and a maximum deviation of 0.0007 for the range 298.1 to 1500 K.

In making the above adjustments Gordon's values of $R(1.9869)$ and of $hc/k(1.4324)$ were used, while the final calculations are to be based on $R = 1.98581$ and $hc/k = 1.43848$. The change in R is made in an obvious manner; that in h/ck is accomplished by multiplying each θ_j by the ratio $1.43848/1.4324 = 1.00424$, leaving the a_j unchanged. Column 3, Table 1, lists the revised values.

The foregoing discussion explains the method used to correct Gordon's data for changes in the physical constants and to interpolate them to close intervals of temperature. There remains only to add the respective corrections of Stephenson and McMahon for the effect of centrifugal distortion, all of which are derivable from the correction to lnQ, namely,

$$1.29 \times 10^{-6} T \ (T \text{ in } ^\circ R) \quad \ldots \quad (5d)$$

Table 2 is a skeleton table giving final values of zero-pressure enthalpy h° and reduced entropy $(s + Rlnp)^\circ$ based on saturated liquid at 32 F as reference point for both enthalpy and entropy.

PROPERTIES OF SATURATED LIQUID

We believe that the most reliable tabulation of properties of liquid water is that of Osborne, Stimson, and Ginnings.[9, 15] These investigators arranged to add energy electrically to an adiabatic calorimeter containing water in both liquid and vapor phases to (α) measure the resulting temperature rise, (β)

TABLE 2—ZERO-PRESSURE PROPERTIES OF WATER VAPOR

t(F)	h^0 BTU/LB	$(s + R\ln p)^0$ BTU/LB R
−288	933.86	1.1586
−272	940.92	1.1979
−256	947.98	1.2340
−240	955.05	1.2674
−224	962.11	1.2984
−208	969.18	1.3274
−192	976.24	1.3546
−176	983.31	1.3803
−160	990.38	1.4045
−144	997.45	1.4275
−128	1004.52	1.4494
−112	1011.59	1.4702
−96	1018.66	1.4901
−80	1025.73	1.5091
−64	1032.81	1.5274
−48	1039.89	1.5449
−32	1046.97	1.5618
−16	1054.06	1.5780
0	1061.15	1.5937
16	1068.24	1.6089
32	1075.34	1.6236
48	1082.45	1.6378
64	1089.56	1.6516
80	1096.68	1.6650
96	1103.81	1.6780
112	1110.95	1.6907
128	1118.10	1.7030
144	1125.26	1.7150
160	1132.43	1.7268
176	1139.62	1.7382
192	1146.81	1.7494
208	1154.02	1.7603
224	1161.25	1.7710

maintain temperature constant by simultaneous withdrawal of saturated liquid, (γ) maintain temperature constant by simultaneous withdrawal of saturated vapor.

Thermodynamic analysis shows that what were measured in the α-experiments are increments of the quantity.

$$\alpha = h_l - Tv_l(dp_s/dT) \quad \ldots \quad (6)$$

for measured increments of international centigrade temperature t. From

the results obtained were derived two empirical equations expressing the dependence of α on temperature t, namely,

$$\alpha = \alpha_0 + 4.1699t + 7.635 \times 10^{-4}t^2 + 0.5634(1 - 10^{-0.101t}) \quad (7a)$$

$$\alpha = \alpha_0 + 0.4858 + 4.17144t \quad (7b)$$

the first being valid in the range 0 to 100 C, the second, in the range 100 to 130 C. The unit of α in each is the international joule per gram (int.joule/g).

Values of the specific volume of the saturated liquid v_f are given with great accuracy in the Bureau of Standards tables, [9, 15] and sufficiently accurate values of the derivative dp_s/dT can be obtained from the unpublished formula of Harold T. Gerry used by the Third International Conference on Steam Tables, namely,

$$\log_{10} p_s = -8.420798(T_s/T - 1) + 8.2 \log_{10} (T_s/T) - 0.92559(1 - T/T_s) \quad (8)$$

where $T = t + 273.16$ and $T_s = 373.16$. Equation (8) will be referred to as the NBS saturation pressure.

Values of the specific enthalpy of the saturated liquid h_f were calculated from the information discussed in the two preceding paragraphs. There is no direct indication in the Bureau of Standards reports [9, 15] whether the unit of α in Equation (7) is the *mean* or the *NBS certified* international joule. Curtis [16] has concluded that the former is larger than the latter by a factor of 1.000026. The difference is small, but is mentioned for the sake of definiteness. We have assumed that the *NBS certified* international joule is intended unless otherwise explicitly stated. The international steam tables kilocalorie has been defined by the factor,

$$860 \text{ kcal/int. kwhr} \quad (9)$$

with the *mean* international joule intended. This has led to a redefinition of the Btu by the factor,

$$1.8 \text{ Btu kg/kcal lb} \quad (10)$$

Accordingly, we compute the factor,

$$0.429989 \text{ Btu} \cdot \text{g/int. (}\textit{NBS certified}\text{) joule} \cdot \text{lb} \quad (11)$$

by which we have converted Bureau of Standards calorimetric data to English units for listing in our final tables.

In the β-experiments were measured values of the quantity,

$$\beta = Tv_f(dp_s/dT) \quad (12)$$

The magnitude of β in the range 0 to 130 C is so small that the values computed in the manner described above are to be regarded as more accurate than those measured directly. The smallness of β also justifies use of the approxi-

mate relation, $T = t + 273.16$, between the absolute and the international temperature scales.

The identical relation,

$$s_t = \tau\beta + \int \tau d\alpha \quad \ldots \ldots \ldots

VAPOR ENTHALPY

Collins and Keyes [6] have performed experiments in which the temperature of steam was restored to its initial value, after throttling to reduced pressure, by adding a measured quantity of energy electrically. In this way they were able to measure the enthalpy at various pressures along an isotherm relative to an arbitrary datum. Analysis by the method of least squares of the observed data yielded values of the temperature coefficients B in the expression,

$$h = h^0 - B_{ww}p - \tfrac{1}{2} B_{www}p^2 \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (16)$$

which were compared with corresponding values computed from the Keyes, Smith, and Gerry [5] (KSG) equation of state. This comparison is exhibited in Table 3. It shows that the KSG values for temperatures below 125 C are not very reliable. Unfortunately, the observed values are not sufficiently extensive to yield, of themselves, a reliable formulation covering the whole range of temperature of interest here. They can be used in conjunction with the extremely accurate Bureau of Standards (NBS) calorimetric data, however, to produce a formulation which appears to be sufficiently reliable for present purposes.

In the Bureau of Standards γ-experiments referred to previously (Properties of Saturated Liquid) the quantity measured is

$$\gamma = Tv_f(dp_s/dT) \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (17)$$

The measurements cover the range of 0 to 374 C at very close temperature intervals. The specific enthalpy of the saturated vapor is

$$h_g = \alpha + \gamma \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (18)$$

hence Equation (16) can be written in the form,

$$B_{ww}p_s + \tfrac{1}{2} B_{www}p_s^2 = h^0 - \alpha - \gamma + C \quad \ldots \ldots \ldots \ldots \quad (19)$$

the constant C being added to allow for the fact that the values of zero-pressure enthalpy h^0 calculated from spectroscopic data are purely relative.

To determine C, the lefthand member of Equation (19) was evaluated at each of the five temperatures in Table 3, using observed values of B_{ww} and B_{www} (except at 38.94 and 59.44 C, where it was necessary to use KSG equation of state values of B_{www} instead), together with NBS saturation pressures Equation (8). The righthand member (except for the constant (C) was evaluated at these same temperatures using our calculated values of zero-pressure enthalpy h^0 and Bureau of Standards values of $\alpha + \gamma$. The constant C was then adjusted to best satisfy Equation (19), the root mean square deviation being only 0.02 Btu per lb. The term $\frac{1}{2} B_{www}p_s^2$ amounts to only 0.07 Btu per lb at 59.44 C and to only 0.02 Btu per lb at 38.94 C; hence the

use of KSG values for B_{www} at these temperatures could not have produced significant error in the adjusted value of C.

The magnitude of $\frac{1}{2} B_{www} p_s^2$ decreases rapidly with decreasing temperature; hence it can be estimated with more than sufficient accuracy in the range 0 to 40 C from KSG equation of state and the NBS saturation pressures. Values of $B_{ww} p_s$ in the range 0 to 40 C were then calculated from Equation (19) using (*a*) the estimated values of $\frac{1}{2} B_{www} p_s^2$; (*b*) our calculated values for zero-pressure enthalpy h^o; (*c*) weighted-mean values of γ given in the Bureau of Standards reports;[9, 15] and (*d*), the adjusted value of the constant

TABLE 3—ENTHALPY COEFFICIENTS, B

t(C)	B_{ww} (INT. JOULE/G ATM)			$\frac{1}{2}B_{www}$ (INT. JOULE/G ATM^2)		
	Observed	KSG	Eq. (20)	Observed	KSG	Eq. (20)
38.94	28.859	33.581	29.345	....	10.395	13.83
59.44	20.774	23.050	20.723	....	4.150	4.93
80.02	15.195	16.616	15.304	2.283	1.910	1.99
100.00	11.871	12.574	11.818	0.832	0.979	0.91
125.00	8.916	9.269	8.903	0.395	0.461	0.38

C. Corresponding values of B_{ww} itself were then obtained by dividing the results by the NBS saturation pressures p_s. We have formulated our final values of B_{ww} and of B_{www} by means of the empirical equations in Equation (20).

$$\left.\begin{aligned} A_{ww} &= -0.0302 + 88.514\tau 10^{232289\tau^2} \text{ (ft}^3\text{/lb)} \\ A_{www} &= 67.4\tau^2 A_{ww}^3 \text{ (ft}^6\text{/lb}^2\text{)} \\ B_{ww} &= -0.0302 + (0.0302 + A_{ww})(2 + 1{,}074{,}300\tau^3) \text{ (ft}^3\text{/lb)} \\ B_{www} &= 202.2\tau^2 A_{ww}^2 B_{ww} \text{ (ft}^6\text{/lb}^2\text{)} \end{aligned}\right\} \quad \ldots . \quad (20)$$

In all of these equations, reciprocal temperature is in $^oR^{-1}$. Values of B_{ww} and of $\frac{1}{2} B_{www}$ calculated from them are listed in Table 3 for comparison with the observations of Collins and Keyes. The agreement appears to be satisfactory.

In Table 4 are listed the NBS weighted-mean values of the calorimetric quantity γ. Deviations from these of the NBS smoothed values and of the values calculated from Equations (19 and 20) are shown for the purpose of comparison.

VAPOR VOLUME

The specific volume of the vapor is to be calculated from the equation,

$$pv = RT - A_{ww} p - A_{www} p^2 \ldots \ldots \ldots \ldots \ldots \ldots \quad (21)$$

TABLE 4—VALUES OF γ (INT.JOULE/G)

t(C)	NBS (WEIGHTED MEAN)	NBS (SMOOTHED)	EQ. (19, 20)
0	2500.72	−0.54	−0.26
1	2497.92	−0.08	0.16
5	2488.34	0.14	0.26
10	2476.73	0.03	0.05
15	2464.81	0.21	0.19
20	2453.36	−0.11	−0.14
25	2441.31	0.15	0.10
30	2429.43	0.21	0.17
40	2405.87	0.01	0.01
50	2381.85	0.08	0.10
60	2357.65	0.11	0.15
70	2333.40	−0.09	−0.06
80	2308.51	0.02	0.04
90	2283.45	−0.10	−0.10
100	2257.73	−0.02	−0.03
Root mean square deviation:		0.0565	0.0452

which is thermodynamically consistent with Equation (16) through the relation Equation (3) provided that

$$\tau A_{ww} = \int B_{ww} d\tau + C_0$$

$$\tau A_{www} = \int B_{www} d\tau + C_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (22)$$

To evaluate the constants C_0 and C_1 requires very accurate determinations of specific volume of at least two different conditions of pressure and temperature. Direct experimental determinations of sufficient accuracy do not seem to be available, at any rate in the range of validity of Equations (20) and (21). However, new and apparently very accurate measurements of saturation pressure in the range 73 to 130 C by Moser and Zmaczynski,[21] together with corresponding values of γ can be used in Equation (17) to compute satisfactory values of specific volume at saturation v_g. Proceeding in this manner, we have found that the constants C_0 and C_1 are both zero within the accuracy of the data. Accordingly, these constants have been omitted in writing the first two of Equation (20).

The first of Equations (20) is based on experimental data in the range 0 to 125 C, the second, on data in the range 80 to 125 C. Undoubtedly extrapolation to higher temperatures is unreliable, but it seems safe to extrapolate to lower temperatures along the saturation curve because the terms $A_{ww}p_s$ and $A_{www}p_s^2$ decrease so rapidly with decreasing temperature. As an estimate of the reliability of our formulation it may be stated that our calculated value of the difference between saturation enthalpy h_g and zero-pressure enthalpy h^o, at 25 C, is 22.2 int.joule/g, whereas the value recommended by Rossini [22] is 22.0 ± 1.0 int.joule/g.

VAPOR PRESSURE OF LIQUID WATER

The only published measurements of the vapor pressure of liquid water (saturation pressure, p_s) at temperatures below 73 C that have come to our attention are the 1917 PTR (Physikalisch-Technische Reichsanstalt) data. Although these have been formulated differently by different authors,[4, 23] they cannot be regarded as being very accurate. Cragoe [24] has kindly shown us his unpublished measurements at 25, 50, 60, and 80 C and these have been compared with our final values in Fig. 1. Because of the smallness of the vapor pressure at low temperatures, it is extremely difficult to measure it accurately there; hence, it seems better to place reliance on values calculated from other data by means of appropriate identical relations of thermodynamics.

Combining (17) and (21) gives the differential equation,

$$\frac{d\ln p_s}{dT} = \frac{\gamma/RT^2}{1 - A_{ww}p_s/RT - A_{www}p_s^2/RT} \quad \ldots\ldots\ldots\ldots\ldots\ldots (23)$$

The negative terms in the denominator are very small compared with unity throughout the range 0 to 100 C; hence, in evaluating them, the use of Equation (20) with NBS saturation pressures shown in Equation (8) is justified. The numerator was evaluated from Equations (19) and (20), and the whole righthand member of Equation (23) was reduced to a semi-empirical equation of suitable form for integration. After some rearrangement and reformulation of the smaller terms, a closed expression for the logarithm of the vapor pressure was obtained, namely, (Equation 24),

$$\begin{aligned}\log_{10}p_s = &-7.90298(T_s/T - 1) + 5.02808\log_{10}(T_s/T)\\ &- 1.3816\cdot10^{-7}\,[10^{11.344(1-T/T_s)} - 1]\\ &+ 8.1328\cdot10^{-3}\,[10^{-3.49149(T_s/T-1)} - 1]\end{aligned} \quad \ldots\ldots\ldots\ldots (24)$$

where p_s is in standard atmospheres and $T_s = 373.16\ K = 671.688\ R$. As a sample calculation, the four terms on the right of Equation (24) are evaluated below for 0C ($T = 273.16\ K$):

$$\begin{aligned}\log_{10}p_s = &-2.893170\\ &+0.681192\\ &-0.000151\\ &-0.007704\\ = &-2.219833;\end{aligned}$$

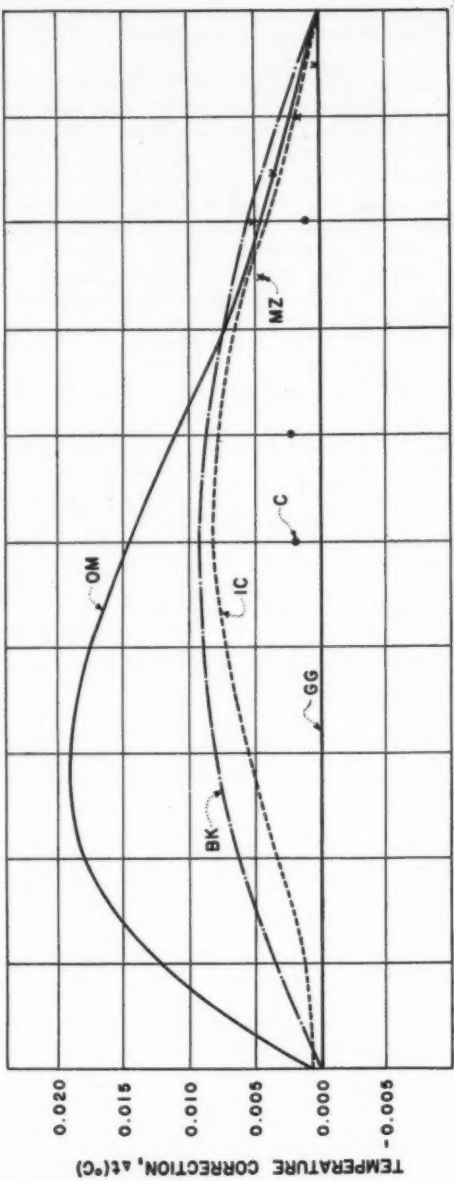
whence

p_s (at 32 F) = 0.0060279 atm.

The estimated uncertainty in $\log_{10}p_s$ as given by Equation (24) is 0.02 per cent. In our final tables the values of p_s itself are given in pounds per square inch (lb/in^2) for which purpose the conversion factor,

$$14.6959\ lb/in^2\ atm \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (25)$$

was used.



GG—Goff-Gratch OM—Osborne-Meyers C—Cragoe MZ—Moser-Zmaczynski IC—Third Int. Conf. on Steam Tables
BK—Blaisdell-Kaye

FIG. 1. COMPARISON OF VAPOR-PRESSURE DATA

Fig. 1 shows an interesting comparison of our results (GG) with those of Osborne and Meyers [23] (*OM*), with those recommended by the Third International Conference on Steam Tables [4] (*IC*), with those of Moser and Zmaczynski [21] (*MZ*), and with those of Cragoe [24] (*C*). The comparison is made in terms of the correction Δt which must be subtracted from the stated temperature t in order to recover the stated vapor pressure p_s on substituting the corrected temperature $t' = t - \Delta t$ into our Equation (24), using the relation $T = t' + 273.16$. The discrepancies between the data of Moser and Zmaczynski and those of Cragoe are especially interesting to note, for they appear to be larger than would be expected considering the degree of precision claimed by these investigators in their measurement of pressure. They can reasonably be attributed, however, to disagreement between the thermometers used, for there is evidence [25] that present specifications of the international centigrade scale are not sufficiently complete or rigid to insure reproducibility to better than ±0.003 C in the range 0 to 100 C.

Blaisdell and Kaye [26] have proposed a centigrade temperature scale t' defined in terms of the resistance R of a suitably constructed platinum thermometer by the cubic,

$$\frac{t'}{100} = \frac{R - R_0}{R_{100} - R_0} + \left(\frac{\delta'}{100}\right)\left(\frac{t'}{100}\right)\left(\frac{t'}{100} - 1\right) + \left(\frac{\gamma'}{t'_s}\right)\left(\frac{t'}{100}\right)\left(\frac{t'}{100} - 1\right)\left(\frac{t'_s}{100} - \frac{t'}{100}\right) \quad (26)$$

where R_0 and R_{100} denote the measured resistance at the ice point and steam point, respectively. The constant δ' is to be evaluated by requiring that $t' = t'_s = 444.7$ when $R = R_s$, the measured resistance at the sulphur point. The constant γ' is to be determined by requiring that $t' = t'_{Hg} = 356.7$ when $R = R_{Hg}$, the measured resistance at the mercury point. The authors state that the constant γ' assumes a value of about -0.035 and that their proposed scale conforms to the absolute scale through the relation, $T = t' + 273.165$, within the experimental error of their measurements. Our Equation (24) expresses the saturation pressure as a function of absolute temperature T.

Most laboratory determinations of the saturation pressure are made at temperatures on the international centigrade scale defined by the quadratic,

$$\frac{t}{100} = \frac{R - R_0}{R_{100} - R_0} + \left(\frac{\delta}{100}\right)\left(\frac{t}{100}\right)\left(\frac{t}{100} - 1\right) \quad (27)$$

where the constant δ is to be evaluated by requiring that $t = 444.6$ when $R = R_s$. Blaisdell and Kaye [26] state that this quadratic gives $t = 356.57$ for $R = R_{Hg}$ so that addition of 273.165 gives a value for the absolute temperature of the mercury point which is too low by 0.13 C. The difference $\Delta t = t - t'$ between the scale proposed by these investigators and the international centigrade scale can be computed at any temperature t from the data given. According to the findings of Blaisdell and Kaye, the value of saturation pressure p_s

calculated from our Equation (24) for $T = t' + 273.16$ should agree with the value measured at temperature t if $t' = t - \Delta t$. Therefore, it is appropriate to plot the computed values of Δt on Fig. 1 also. The resulting curve is marked (BK).

At first it would appear that Fig. 1 contains evidence supporting the findings of Blaisdell and Kaye regarding the true relation between the international and absolute temperature scales. No definite conclusions can be drawn, however, because the corrections Δt appear to be well within the uncertainties of the experimental values.

VAPOR PRESSURE OF ICE

The vapor pressure of ice has been calculated recently by Goff.[27] Harrison [28] has compared his values with the few experimental data available and found good agreement. In preceding paragraphs some of the data on which Goff based his calculations have been revised so that it appears desirable, in the interest of consistency, to revise the calculations also.

The differential equation to be used is the same as Equation (23) except that the enthalpy of sublimation h_{ig} replaces γ in the numerator on the right. Strictly speaking, the quantity $p_s v_l / RT$ should be subtracted from the denominator, but its magnitude is so very small compared with unity that no appreciable error is committed in neglecting to do so. From Equation (16),

$$h_{ig} = h^o - h_l - B_{ww} p_s - \tfrac{1}{2} B_{www} p_s^2$$

The difference, $h^o - h_l$, between zero-pressure enthalpy of the vapor and saturation enthalpy of the solid is a large quantity which varies only slightly with temperature. Its value at 0C (32 F) is 1218.74 Btu per lb, according to the present analysis; hence, we may write

$$h^o - h_l = 1218.74 + \Delta h$$

where Δh is a small quantity which is readily evaluated from data discussed in preceding paragraphs. Letting p_o denote the value of the vapor pressure at 0C, we obtain,

$$\log_{10}(p_s/p_o) = -9.76670(T_o/T - 1)$$

$$+0.43429 \int_{To}^{T} \frac{\Delta h/RT - B_{ww} p_s/RT - \tfrac{1}{2} B_{www} p_s^2/RT}{1 - A_{ww} p_s/RT - A_{www} p_s^2/RT} \, dT/T$$

$$+0.43429 \int_{To}^{T} \frac{(1218.74/RT)(A_{ww} p_s/RT + A_{www} p_s^2/RT)}{1 - A_{ww} p_s/RT - A_{www} p_s^2/RT} \, dT/T. \quad (28)$$

where $T_o = 273.16K = 491.688R$.

The results of the above recalculation agree with those previously published well within their estimated uncertainty, namely, ±0.03 per cent in $\log_{10}(p_s/p_o)$. The recalculations are summarized in Table 5.

To obtain the vapor pressure p_s itself from the data summarized in Table 5 requires to know its value at 0C, namely p_o. According to International Critical Tables,[29] the triple point of water is at 32.0176 F. At this point the vapor pressure of the solid is identical with that of the liquid. At 32 F the difference between them has been estimated[2] as 0.0000006 atm. But from

TABLE 5—RECALCULATED VALUES OF THE VAPOR PRESSURE OF ICE

t(F)	$\log_{10}(p_s/p_o)$	t(F)	$\log_{10}(p_s/p_o)$	t(F)	$\log_{10}(p_s/p_o)$
32	0.00000	−80	$\bar{3}.11588$	−192	$\bar{9}.84521$
16	$\bar{1}.67130$	−96	$\bar{4}.55943$	−208	$\bar{10}.71610$
0	$\bar{1}.31957$	−112	$\bar{5}.95227$	−224	$\bar{11}.43649$
−16	$\bar{2}.94235$	−128	$\bar{5}.28723$	−240	$\bar{13}.97398$
−32	$\bar{2}.53686$	−144	$\bar{6}.55570$	−256	$\bar{14}.28606$
−48	$\bar{2}.09987$	−160	$\bar{7}.74721$	−272	$\bar{16}.31581$
−64	$\bar{3}.62766$	−176	$\bar{8}.84898$	−288	$\bar{19}.98529$

Equation (24) the vapor pressure of the liquid at 32 F is 0.0060279 atm; accordingly, that of the solid at 32 F is

$$p_o = 0.0060273 \text{ atm.} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (29)$$

VAPOR ENTROPY

From Equation (21) can be derived an expression for reduced entropy, namely,

$$s + R\ln p = s^0 + C_{ww}p + \tfrac{1}{2}C_{www}p^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots (30)$$

where

$$C_{ww} = \tau A_{ww} - \tau B_{ww}$$
$$C_{www} = \tau A_{www} - \tau B_{www} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (31)$$

The method of calculating zero-pressure reduced entropy s^0 has been explained in the section on Zero-Pressure Properties of Water Vapor. The calculated values are purely relative; hence, it is necessary to add a constant adjusted to make $s_t = 0$ at 32 F. Since it is also desired to make $h_t = 0$ at 32 F, in conformity with steam table practice, the adjustment is accomplished by

TABLE 6. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION: SOLID-VAPOR

Temp. Fahr. t	Abs. Pressure P_s lb/sq in	Abs. Pressure P_s in Hg	Specific Volume, cu ft/lb Sat.Solid v_i	Evap. $v_{ig} \times 10^{-8}$	Sat.Vapor $v_g \times 10^{-3}$	Enthalpy, Btu/lb Sat.Solid h_i	Evap. h_{ig}	Sat.Vapor h_g	Entropy, Btu/lb°F Sat.Solid s_i	Evap. s_{ig}	Sat.Vapor s_g	Temp. Fahr. t
32	0.08988	0.1803	0.01747	3.306	3.306	-143.40	1218.56	1075.16	-0.3916	2.4785	2.1867	32
31	0.08461	0.1723	0.01747	3.453	3.453	-143.90	1218.63	1074.73	-0.3937	2.4836	2.1900	31
30	0.08080	0.1645	0.01747	3.608	3.608	-144.40	1218.69	1074.29	-0.3937	2.4887	2.1950	30
29	0.07715	0.1571	0.01747	3.771	3.771	-144.90	1218.75	1073.86	-0.3947	2.4939	2.1992	29
28	0.07368	0.1500	0.01746	3.943	3.943	-145.40	1218.81	1073.41	-0.3957	2.4991	2.2034	28
27	0.07030	0.1431	0.01746	4.122	4.122	-145.90	1218.87	1072.97	-0.3968	2.5045	2.2077	27
26	0.06708	0.1366	0.01746	4.311	4.311	-146.40	1218.93	1072.53	-0.3978	2.5097	2.2119	26
25	0.06400	0.1303	0.01746	4.509	4.509	-146.89	1218.98	1072.09	-0.3988	2.5150	2.2162	25
24	0.06108	0.1243	0.01746	4.717	4.717	-147.39	1219.05	1071.66	-0.3998	2.5203	2.2205	24
23	0.05823	0.1186	0.01746	4.936	4.936	-147.88	1219.10	1071.22	-0.3008	2.5256	2.2249	23
22	0.05552	0.1130	0.01746	5.166	5.166	-148.38	1219.16	1070.78	-0.3019	2.5310	2.2291	22
21	0.05293	0.1078	0.01745	5.408	5.408	-148.87	1219.21	1070.34	-0.3029	2.5364	2.2335	21
20	0.05045	0.1027	0.01745	5.662	5.662	-149.36	1219.26	1069.90	-0.3039	2.5417	2.2378	20
19	0.04809	0.09789	0.01745	5.929	5.929	-149.86	1219.31	1069.46	-0.3049	2.5471	2.2422	19
18	0.04581	0.09326	0.01745	6.210	6.210	-150.34	1219.36	1069.02	-0.3060	2.5526	2.2466	18
17	0.04363	0.08884	0.01745	6.506	6.506	-150.83	1219.41	1068.58	-0.3070	2.5581	2.2511	17
16	0.04154	0.08461	0.01745	6.817	6.817	-151.32	1219.46	1068.14	-0.3080	2.5635	2.2555	16
15	0.03957	0.08056	0.01744	7.144	7.144	-151.80	1219.50	1067.70	-0.3090	2.5690	2.2600	15
14	0.03767	0.07669	0.01744	7.489	7.489	-152.29	1219.55	1067.26	-0.3101	2.5746	2.2645	14
13	0.03585	0.07300	0.01744	7.851	7.851	-152.77	1219.59	1066.82	-0.3111	2.5801	2.2690	13
12	0.03412	0.06946	0.01744	8.234	8.234	-153.26	1219.64	1066.38	-0.3121	2.5857	2.2736	12
11	0.03246	0.06609	0.01744	8.636	8.636	-153.74	1219.68	1065.94	-0.3131	2.5912	2.2781	11
10	0.03087	0.06286	0.01744	9.060	9.060	-154.22	1219.72	1065.50	-0.3142	2.5969	2.2827	10
9	0.02936	0.05977	0.01744	9.507	9.507	-154.70	1219.76	1065.06	-0.3152	2.6025	2.2873	9
8	0.02791	0.05683	0.01743	9.979	9.979	-155.18	1219.80	1064.62	-0.3162	2.6081	2.2919	8
7	0.02653	0.05402	0.01743	10.48	10.48	-155.66	1219.84	1064.18	-0.3172	2.6139	2.2966	7
6	0.02521	0.05134	0.01743	11.00	11.00	-156.14	1219.88	1063.74	-0.3182	2.6194	2.3012	6

Temp. Fahr. t	Abs. Pressure $P_s \times 10^3$ lb/sq in	Abs. Pressure $P_s \times 10^3$ in Hg	Specific Volume, cu ft/lb Sat.Solid v_i	Evap. $v_{ig} \times 10^{-6}$	Sat.Vapor $v_g \times 10^{-6}$	Enthalpy, Btu/lb Sat.Solid h_i	Evap. h_{ig}	Sat.Vapor h_g	Entropy, Btu/lb°F Sat.Solid s_i	Evap. s_{ig}	Sat.vapor s_g	Temp. Fahr. t
5	2.396	4.878	0.01743	1.155	1.155	-156.61	1219.90	1063.29	-0.3193	2.6252	2.3059	5
4	2.276	4.633	0.01743	1.214	1.214	-157.09	1219.94	1062.85	-0.3203	2.6309	2.3106	4
3	2.161	4.400	0.01743	1.275	1.275	-157.56	1219.97	1062.41	-0.3213	2.6367	2.3154	3
2	2.052	4.178	0.01742	1.340	1.340	-158.04	1220.01	1061.97	-0.3224	2.6425	2.3201	2
1	1.948	3.966	0.01742	1.408	1.408	-158.51	1220.04	1061.53	-0.3234	2.6483	2.3249	1
0	1.849	3.764	0.01742	1.481	1.481	-158.98	1220.07	1061.09	-0.3244	2.6541	2.3297	0
-1	1.754	3.572	0.01742	1.557	1.557	-159.45	1220.10	1060.65	-0.3254	2.6600	2.3346	-1
-2	1.664	3.388	0.01742	1.638	1.638	-159.92	1220.13	1060.21	-0.3264	2.6658	2.3394	-2
-3	1.578	3.213	0.01742	1.723	1.723	-160.39	1220.15	1059.76	-0.3275	2.6718	2.3443	-3
-4	1.496	3.047	0.01742	1.814	1.814	-160.86	1220.18	1059.32	-0.3285	2.6777	2.3492	-4
-5	1.419	2.888	0.01741	1.909	1.909	-161.33	1220.21	1058.88	-0.3295	2.6836	2.3541	-5
-6	1.344	2.737	0.01741	2.010	2.010	-161.79	1220.23	1058.44	-0.3306	2.6896	2.3590	-6
-7	1.274	2.594	0.01741	2.116	2.116	-162.26	1220.26	1058.00	-0.3316	2.6956	2.3640	-7
-8	1.207	2.457	0.01741	2.229	2.229	-162.72	1220.28	1057.56	-0.3326	2.7016	2.3690	-8
-9	1.143	2.327	0.01741	2.349	2.349	-163.18	1220.30	1057.12	-0.3336	2.7076	2.3740	-9
-10	1.082	2.203	0.01741	2.475	2.475	-163.65	1220.32	1056.67	-0.3347	2.7138	2.3791	-10
-11	1.024	2.086	0.01740	2.609	2.609	-164.11	1220.34	1056.23	-0.3357	2.7198	2.3841	-11
-12	0.9694	1.974	0.01740	2.750	2.750	-164.57	1220.36	1055.79	-0.3367	2.7259	2.3892	-12
-13	0.9172	1.867	0.01740	2.901	2.901	-165.03	1220.38	1055.35	-0.3377	2.7320	2.3943	-13
-14	0.8676	1.766	0.01740	3.060	3.060	-165.48	1220.39	1054.91	-0.3388	2.7383	2.3995	-14
-15	0.8204	1.670	0.01740	3.228	3.228	-165.94	1220.41	1054.47	-0.3398	2.7444	2.4046	-15
-16	0.7757	1.579	0.01740	3.407	3.407	-166.40	1220.42	1054.02	-0.3408	2.7506	2.4098	-16
-17	0.7332	1.493	0.01740	3.596	3.596	-166.86	1220.43	1053.58	-0.3418	2.7568	2.4150	-17
-18	0.6929	1.410	0.01740	3.797	3.797	-167.31	1220.45	1053.14	-0.3429	2.7632	2.4203	-18
-19	0.6545	1.333	0.01739	4.011	4.011	-167.76	1220.46	1052.70	-0.3439	2.7695	2.4256	-19
-20	0.6181	1.259	0.01739	4.237	4.237	-168.21	1220.47	1052.26	-0.3449	2.7757	2.4308	-20
-21	0.5836	1.188	0.01739	4.477	4.477	-168.66	1220.48	1051.82	-0.3460	2.7822	2.4362	-21
-22	0.5509	1.122	0.01739	4.732	4.732	-169.12	1220.49	1051.37	-0.3470	2.7885	2.4415	-22
-23	0.5199	1.059	0.01739	5.003	5.003	-169.56	1220.49	1050.93	-0.3480	2.7949	2.4469	-23
-24	0.4905	0.9987	0.01739	5.290	5.290	-170.01	1220.50	1050.49	-0.3490	2.8013	2.4523	-24

TABLE 6. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION: SOLID-VAPOR *(Continued)*

Temp. Fahr. t	Abs. Pressure $p_a \times 10^3$ Lb/sq in	Abs. Pressure $p_a \times 10^3$ In Hg	Specific Volume, cu ft/lb Sat. Solid v_i	Evap. $v_{ig} \times 10^{-5}$	Sat. Vapor $v_g \times 10^{-5}$	Enthalpy, Btu/lb Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Entropy, Btu/lb°F Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	Temp. Fahr. t
-85	4.687	9.480	0.01739	0.5696	0.5696	-170.46	1220.51	1050.05	-0.3801	2.8078	2.4277	-85
-86	4.563	9.229	0.01738	0.5921	0.5921	-170.91	1220.81	1049.90	-0.3811	2.8143	2.4332	-86
-87	4.113	8.318	0.01738	0.6367	0.6367	-171.35	1220.81	1049.16	-0.3821	2.8207	2.4386	-87
-88	3.976	7.891	0.01738	0.6634	0.6634	-171.80	1220.52	1048.72	-0.3831	2.8274	2.4743	-88
-89	3.858	7.635	0.01738	0.7085	0.7085	-172.24	1220.52	1048.28	-0.3842	2.8339	2.4797	-89
-90	3.440	7.003	0.01738	0.7441	0.7441	-172.69	1220.52	1047.94	-0.3852	2.8405	2.4853	-90
-91	3.839	6.595	0.01738	0.7883	0.7883	-173.13	1220.53	1047.40	-0.3862	2.8470	2.4908	-91
-92	3.049	6.308	0.01738	0.8355	0.8355	-173.57	1220.52	1046.95	-0.3873	2.8538	2.4965	-92
-93	2.890	5.843	0.01738	0.8857	0.8857	-174.01	1220.52	1046.51	-0.3883	2.8604	2.5021	-93
-94	2.700	5.497	0.01737	0.9391	0.9391	-174.45	1220.52	1046.07	-0.3893	2.8671	2.5078	-94
-95	2.540	5.190	0.01737	0.9961	0.9961	-174.88	1220.51	1045.63	-0.3904	2.8739	2.5135	-95
-96	2.388	4.962	0.01737	1.057	1.057	-175.32	1220.51	1045.19	-0.3914	2.8807	2.5193	-96
-97	2.245	4.570	0.01737	1.122	1.122	-175.76	1220.50	1044.74	-0.3924	2.8874	2.5250	-97
-98	2.110	4.296	0.01737	1.191	1.191	-176.19	1220.49	1044.30	-0.3934	2.8942	2.5308	-98
-99	1.982	4.035	0.01737	1.264	1.264	-176.63	1220.49	1043.86	-0.3945	2.9012	2.5367	-99
-100	1.861	3.790	0.01737	1.343	1.343	-177.06	1220.48	1043.42	-0.3955	2.9080	2.5425	-100
-101	1.746	3.559	0.01736	1.427	1.427	-177.49	1220.47	1042.98	-0.3965	2.9149	2.5484	-101
-102	1.641	3.340	0.01736	1.516	1.516	-177.93	1220.45	1042.53	-0.3976	2.9219	2.5543	-102
-103	1.539	3.134	0.01736	1.612	1.612	-178.36	1220.44	1042.09	-0.3986	2.9289	2.5603	-103
-104	1.444	2.940	0.01736	1.715	1.715	-178.79	1220.43	1041.65	-0.3996	2.9359	2.5663	-104
-105	1.354	2.757	0.01736	1.824	1.824	-179.21	1220.42	1041.21	-0.4007	2.9430	2.5723	-105
-106	1.270	2.585	0.01736	1.941	1.941	-179.64	1220.40	1040.76	-0.4017	2.9501	2.5784	-106
-107	1.190	2.422	0.01736	2.066	2.066	-180.06	1220.38	1040.32	-0.4027	2.9571	2.5844	-107
-108	1.115	2.270	0.01736	2.200	2.200	-180.49	1220.37	1039.88	-0.4038	2.9643	2.5905	-108
-109	1.044	2.126	0.01735	2.343	2.343	-180.92	1220.36	1039.44	-0.4048	2.9715	2.5967	-109
-110	0.9776	1.990	0.01735	2.496	2.496	-181.34	1220.34	1039.00	-0.4058	2.9786	2.6028	-110
-111	0.9151	1.863	0.01735	2.660	2.660	-181.76	1220.31	1038.55	-0.4069	2.9860	2.6091	-111
-112	0.8568	1.743	0.01735	2.836	2.836	-182.18	1220.29	1038.11	-0.4079	2.9933	2.6153	-112
-113	0.8009	1.631	0.01735	3.024	3.024	-182.60	1220.27	1037.67	-0.4089	3.0008	2.6218	-113
-114	0.7489	1.525	0.01735	3.226	3.226	-183.02	1220.25	1037.23	-0.4100	3.0079	2.6279	-114

Temp. Fahr. t	Abs. Pressure $p_a \times 10^4$ Lb/sq in	Abs. Pressure $p_a \times 10^4$ In Hg	Specific Volume, cu ft/lb Sat. Solid v_i	Evap. $v_{ig} \times 10^{-5}$	Sat. Vapor $v_g \times 10^{-5}$	Enthalpy, Btu/lb Sat. Solid $-h_i$	Evap. h_{ig}	Sat. Vapor h_g	Entropy, Btu/lb°F Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	Temp. Fahr. t
-55	7.001	14.26	0.01734	3.443	3.443	-183.44	1220.23	1036.79	-0.3810	3.0152	2.6342	-55
-56	6.543	13.32	0.01734	3.675	3.675	-183.86	1220.20	1036.34	-0.3820	3.0226	2.6406	-56
-57	6.118	12.44	0.01734	3.925	3.925	-184.28	1220.18	1035.90	-0.3831	3.0301	2.6470	-57
-58	5.708	11.62	0.01734	4.192	4.192	-184.69	1220.15	1035.46	-0.3841	3.0376	2.6535	-58
-59	5.323	10.83	0.01734	4.479	4.479	-185.11	1220.13	1035.02	-0.3851	3.0450	2.6599	-59
-60	4.972	10.12	0.01734	4.786	4.786	-185.52	1220.10	1034.58	-0.3862	3.0526	2.6664	-60
-61	4.639	9.444	0.01734	5.120	5.120	-185.93	1220.06	1034.13	-0.3872	3.0602	2.6730	-61
-62	4.326	8.806	0.01734	5.476	5.476	-186.35	1220.04	1033.69	-0.3882	3.0679	2.6796	-62
-63	4.033	8.211	0.01733	5.859	5.859	-186.76	1220.01	1033.25	-0.3893	3.0756	2.6862	-63
-64	3.758	7.652	0.01733	6.272	6.272	-187.17	1219.98	1032.81	-0.3903	3.0834	2.6929	-64
-65	3.501	7.129	0.01733	6.715	6.715	-187.58	1219.94	1032.36	-0.3914	3.0910	2.6996	-65
-66	3.260	6.638	0.01733	7.193	7.193	-187.98	1219.90	1031.92	-0.3924	3.0987	2.7063	-66
-67	3.035	6.180	0.01733	7.707	7.707	-188.39	1219.87	1031.48	-0.3934	3.1064	2.7130	-67
-68	2.824	5.751	0.01733	8.261	8.261	-188.80	1219.84	1031.04	-0.3945	3.1143	2.7198	-68
-69	2.627	5.349	0.01732	8.856	8.856	-189.20	1219.80	1030.60	-0.3955	3.1222	2.7267	-69
-70	2.443	4.974	0.01732	9.501	9.501	-189.61	1219.76	1030.15	-0.3965	3.1301	2.7336	-70
-71	2.271	4.624	0.01732	10.20	10.20	-190.01	1219.72	1029.71	-0.3976	3.1381	2.7405	-71
-72	2.110	4.296	0.01732	10.94	10.94	-190.41	1219.68	1029.27	-0.3986	3.1460	2.7474	-72
-73	1.960	3.991	0.01732	11.75	11.75	-190.82	1219.65	1028.83	-0.3996	3.1540	2.7544	-73
-74	1.820	3.705	0.01732	12.62	12.62	-191.22	1219.60	1028.38	-0.4007	3.1622	2.7615	-74
-75	1.689	3.439	0.01732	13.54	13.54	-191.62	1219.56	1027.94	-0.4017	3.1702	2.7686	-75
-76	1.567	3.191	0.01732	14.53	14.53	-192.01	1219.51	1027.50	-0.4028	3.1784	2.7756	-76
-77	1.453	2.959	0.01732	15.58	15.58	-192.41	1219.47	1027.06	-0.4038	3.1866	2.7828	-77
-78	1.347	2.743	0.01731	16.67	16.67	-192.81	1219.43	1026.62	-0.4049	3.1948	2.7900	-78
-79	1.249	2.542	0.01731	18.16	18.16	-193.20	1219.37	1026.17	-0.4058	3.2030	2.7972	-79
-80	1.157	2.356	0.01731	19.55	19.55	-193.60	1219.33	1025.73	-0.4069	3.2114	2.8045	-80
-81	1.071	2.181	0.01731	21.06	21.06	-193.99	1219.29	1025.29	-0.4079	3.2197	2.8118	-81
-82	0.9913	2.018	0.01731	22.70	22.70	-194.38	1219.23	1024.85	-0.4090	3.2282	2.8192	-82
-83	0.9171	1.867	0.01731	24.46	24.46	-194.78	1219.19	1024.40	-0.4100	3.2366	2.8266	-83
-84	0.8488	1.727	0.01731	26.36	26.36	-195.16	1219.12	1023.96	-0.4110	3.2450	2.8340	-84

TABLE 6. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION: SOLID-VAPOR *(Continued)*

Temp. Fahr. t	Abs. Pressure $p_s \times 10^3$ Lb/Sq In	Abs. Pressure $p_s \times 10^3$ In Hg	Specific Volume, cu ft/lb Sat.Solid v_1	Specific Volume, cu ft/lb Evap. $v_{lg} \times 10^{-4}$	Specific Volume, cu ft/lb Sat.Vapor $v_g \times 10^{-4}$	Enthalpy, Btu/lb Sat.Solid h_1	Enthalpy, Btu/lb Evap. h_{lg}	Enthalpy, Btu/lb Sat.Vapor h_g	Entropy, Btu/lb°F Sat.Solid s_1	Entropy, Btu/lb°F Evap. s_{lg}	Entropy, Btu/lb°F Sat.Vapor s_g	Temp. Fahr. t
-85	7.841	15.96	0.01730	2.844	2.844	-196.84	1219.08	1083.58	-0.4181	3.8236	3.8415	-85
-86	7.540	14.79	0.01730	2.972	2.972	-196.94	1219.08	1083.08	-0.4131	3.8481	3.8490	-86
-87	6.998	13.63	0.01730	3.317	3.317	-196.99	1218.97	1082.64	-0.4148	3.8706	3.8564	-87
-88	6.179	12.58	0.01730	3.693	3.693	-196.78	1218.98	1082.90	-0.4152	3.8794	3.8643	-88
-89	5.768	11.61	0.01730	3.872	3.872	-197.10	1218.88	1081.78	-0.4168	3.8890	3.8718	-89
-90	5.380	10.71	0.01730	4.186	4.186	-197.49	1218.80	1081.31	-0.4173	3.8969	3.8796	-90
-91	4.850	9.874	0.01730	4.525	4.525	-197.97	1218.74	1080.87	-0.4183	3.9064	3.8873	-91
-92	4.470	9.108	0.01730	4.899	4.899	-198.85	1218.68	1080.43	-0.4194	3.9145	3.8951	-92
-93	4.118	8.388	0.01730	5.303	5.303	-198.64	1218.68	1019.98	-0.4804	3.8833	3.9029	-93
-94	3.793	7.723	0.01729	5.743	5.743	-199.08	1218.56	1019.54	-0.4814	3.8922	3.9108	-94
-95	3.491	7.109	0.01729	6.223	6.223	-199.40	1218.50	1019.10	-0.4825	3.9418	3.9187	-95
-96	3.213	6.539	0.01729	6.745	6.745	-199.78	1218.44	1018.66	-0.4835	3.9508	3.9267	-96
-97	2.954	6.014	0.01729	7.314	7.314	-200.15	1218.37	1018.22	-0.4846	3.9593	3.9347	-97
-98	2.715	5.528	0.01729	7.936	7.936	-200.53	1218.30	1017.77	-0.4856	3.9684	3.9428	-98
-99	2.494	5.079	0.01729	8.613	8.613	-200.90	1218.23	1017.33	-0.4866	3.9775	3.9509	-99
-100	2.291	4.664	0.01729	9.358	9.358	-201.29	1218.17	1016.89	-0.4877	3.9866	3.9601	-100
-101	2.103	4.281	0.01729	10.16	10.16	-201.65	1218.10	1016.45	-0.4887	3.9960	3.9673	-101
-102	1.929	3.929	0.01728	11.04	11.04	-202.02	1218.04	1016.01	-0.4898	3.9053	3.9755	-102
-103	1.769	3.602	0.01728	12.01	12.01	-202.40	1217.96	1015.56	-0.4908	3.9146	3.9838	-103
-104	1.622	3.301	0.01728	13.07	13.07	-202.77	1217.89	1015.12	-0.4919	3.9240	3.9922	-104
-105	1.486	3.025	0.01728	14.22	14.22	-203.14	1217.82	1014.68	-0.4929	3.9336	3.0006	-105
-106	1.360	2.770	0.01728	15.49	15.49	-203.51	1217.75	1014.24	-0.4939	3.9430	3.0091	-106
-107	1.245	2.535	0.01728	16.87	16.87	-203.88	1217.68	1013.80	-0.4950	3.9526	3.0176	-107
-108	1.139	2.319	0.01728	18.39	18.39	-204.24	1217.60	1013.36	-0.4960	3.9621	3.0261	-108
-109	1.041	2.120	0.01728	20.06	20.06	-204.61	1217.52	1012.91	-0.4970	3.9718	3.0348	-109
-110	0.9517	1.938	0.01728	21.89	21.89	-204.98	1217.45	1012.47	-0.4981	3.9812	3.0434	-110
-111	0.8693	1.770	0.01727	23.89	23.89	-205.34	1217.37	1012.03	-0.4991	3.9918	3.0521	-111
-112	0.7936	1.616	0.01727	26.10	26.10	-205.70	1217.29	1011.59	-0.4998	3.9011	3.0609	-112
-113	0.7241	1.474	0.01727	28.52	28.52	-206.07	1217.21	1011.14	-0.4418	3.8109	3.0697	-113
-114	0.6604	1.344	0.01727	31.18	31.18	-206.43	1217.13	1010.70	-0.4423	3.8209	3.0786	-114

Temp. Fahr. t	Abs. Pressure $p_s \times 10^3$ Lb/Sq In	Abs. Pressure $p_s \times 10^3$ In Hg	Specific Volume, cu ft/lb Sat.Solid v_1	Specific Volume, cu ft/lb Evap. $v_{lg} \times 10^{-7}$	Specific Volume, cu ft/lb Sat.Vapor $v_g \times 10^{-7}$	Enthalpy, Btu/lb Sat.Solid h_1	Enthalpy, Btu/lb Evap. h_{lg}	Enthalpy, Btu/lb Sat.Vapor h_g	Entropy, Btu/lb°F Sat.Solid s_1	Entropy, Btu/lb°F Evap. s_{lg}	Entropy, Btu/lb°F Sat.Vapor s_g	Temp. Fahr. t
-115	6.019	12.24	0.01727	3.411	3.411	-206.79	1217.06	1010.26	-0.4435	3.8308	3.0873	-115
-116	5.494	11.16	0.01727	3.733	3.733	-207.15	1216.97	1009.82	-0.4444	3.8409	3.0965	-116
-117	4.993	10.17	0.01727	4.088	4.088	-207.51	1216.89	1009.38	-0.4454	3.8510	3.1056	-117
-118	4.544	9.231	0.01727	4.480	4.480	-207.86	1216.80	1008.94	-0.4464	3.8611	3.1147	-118
-119	4.138	8.416	0.01726	4.911	4.911	-208.22	1216.71	1008.49	-0.4475	3.8713	3.1239	-119
-120	3.767	7.649	0.01726	5.386	5.386	-208.58	1216.63	1008.05	-0.4485	3.8815	3.1330	-120
-121	3.413	6.949	0.01726	5.911	5.911	-208.93	1216.54	1007.61	-0.4496	3.8919	3.1423	-121
-122	3.099	6.310	0.01726	6.491	6.491	-209.30	1216.45	1007.17	-0.4506	3.8022	3.1516	-122
-123	2.812	5.726	0.01726	7.131	7.131	-209.64	1216.37	1006.73	-0.4517	3.8127	3.1610	-123
-124	2.551	5.194	0.01725	7.839	7.839	-209.99	1216.27	1006.28	-0.4527	3.8232	3.1705	-124
-125	2.313	4.709	0.01724	8.622	8.622	-210.34	1216.19	1005.84	-0.4538	3.8338	3.1800	-125
-126	2.096	4.268	0.01724	9.489	9.489	-210.69	1216.09	1005.40	-0.4549	3.8444	3.1896	-126
-127	1.897	3.868	0.01724	10.45	10.45	-211.04	1216.00	1004.96	-0.4559	3.8551	3.1992	-127
-128	1.716	3.494	0.01724	11.51	11.51	-211.39	1215.91	1004.52	-0.4569	3.8658	3.2089	-128
-129	1.552	3.160	0.01725	12.69	12.69	-211.73	1215.80	1004.07	-0.4580	3.8766	3.2186	-129
-130	1.403	2.856	0.01725	14.00	14.00	-212.08	1215.71	1003.63	-0.4590	3.8874	3.2284	-130
-131	1.267	2.579	0.01725	15.45	15.45	-212.43	1215.62	1003.19	-0.4601	3.8984	3.2383	-131
-132	1.144	2.329	0.01725	17.07	17.07	-212.77	1215.52	1002.75	-0.4611	3.9093	3.2482	-132
-133	1.032	2.101	0.01725	18.84	18.84	-213.11	1215.42	1002.31	-0.4622	3.9205	3.2583	-133
-134	0.9301	1.894	0.01725	20.86	20.86	-213.46	1215.32	1001.86	-0.4632	3.9315	3.2683	-134
-135	0.8380	1.706	0.01725	23.08	23.08	-213.80	1215.22	1001.42	-0.4643	3.9428	3.2785	-135
-136	0.7546	1.536	0.01725	25.55	25.55	-214.14	1215.12	1000.98	-0.4653	3.9540	3.2887	-136
-137	0.6790	1.382	0.01724	28.31	28.31	-214.48	1215.02	1000.54	-0.4664	3.9653	3.2989	-137
-138	0.6106	1.243	0.01724	31.38	31.38	-214.82	1214.92	1000.10	-0.4674	3.9766	3.3092	-138
-139	0.5497	1.117	0.01724	34.81	34.81	-215.16	1214.82	999.66	-0.4685	3.9881	3.3196	-139
-140	0.4958	1.003	0.01724	38.64	38.64	-215.49	1214.70	999.21	-0.4695	3.9996	3.3301	-140
-141	0.4483	0.9008	0.01724	42.92	42.92	-215.83	1214.60	998.77	-0.4706	3.0112	3.3406	-141
-142	0.3967	0.8074	0.01724	47.70	47.70	-216.16	1214.49	998.33	-0.4716	3.0229	3.3513	-142
-143	0.3606	0.7240	0.01724	53.08	53.08	-216.50	1214.39	997.88	-0.4727	3.0346	3.3619	-143
-144	0.3194	0.6483	0.01724	59.08	59.08	-216.83	1214.28	997.45	-0.4737	3.0464	3.3727	-144

TABLE 6. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION: SOLID-VAPOR *(Concluded)*

Temp. Fahr. t	Abs. Pressure $p_s \times 10^9$ lb/sq in	in Hg	Specific Volume, cu ft/lb Sat. Solid v_i	Evap. $v_{ig} \times 10^{-6}$	Sat. Vapor $v_g \times 10^{-6}$	Enthalpy, Btu/lb Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Entropy, Btu/lb°F Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	Temp. Fahr. t
-145	2.880	5.803	0.01784	6.577	6.577	-217.17	1214.17	997.00	-0.4748	3.8563	3.3835	-145
-146	2.549	5.190	0.01784	7.330	7.330	-217.50	1214.06	996.56	-0.4756	3.8708	3.3944	-146
-147	2.278	4.639	0.01784	8.174	8.174	-217.82	1213.95	996.13	-0.4769	3.8823	3.4054	-147
-148	2.055	4.143	0.01784	9.123	9.123	-218.14	1213.84	995.68	-0.4780	3.8944	3.4164	-148
-149	1.816	3.698	0.01783	10.19	10.19	-218.49	1213.73	995.24	-0.4790	3.9065	3.4275	-149
-150	1.620	3.298	0.01783	11.39	11.39	-218.82	1213.62	994.80	-0.4801	3.9186	3.4387	-150
-151	1.444	2.939	0.01783	12.74	12.74	-219.14	1213.49	994.35	-0.4811	3.9311	3.4500	-151
-152	1.286	2.618	0.01783	14.26	14.26	-219.47	1213.36	993.91	-0.4822	3.9435	3.4613	-152
-153	1.144	2.329	0.01783	15.97	15.97	-219.79	1213.24	993.47	-0.4832	3.9559	3.4727	-153
-154	1.017	2.072	0.01783	17.90	17.90	-220.13	1213.15	993.03	-0.4843	3.9685	3.4842	-154
-155	0.9040	1.840	0.01783	20.08	20.08	-220.44	1213.02	992.58	-0.4854	3.9812	3.4958	-155
-156	0.8028	1.634	0.01783	22.54	22.54	-220.76	1212.90	992.14	-0.4864	3.9939	3.5075	-156
-157	0.7131	1.450	0.01783	25.32	25.32	-221.09	1212.79	991.70	-0.4875	4.0067	3.5192	-157
-158	0.6318	1.285	0.01783	28.47	28.47	-221.41	1212.67	991.26	-0.4886	4.0196	3.5310	-158
-159	0.5598	1.139	0.01782	32.03	32.03	-221.73	1212.55	990.82	-0.4896	4.0325	3.5429	-159
-160	0.4949	1.008	0.01782	36.07	36.07	-222.05	1212.43	990.38	-0.4907	4.0456	3.5549	-160

making s_g calculated from Equation (30) for $p = p_s$ equal to h_g/T calculated from Equation (16), both at 32 F, of course.

NUMERICAL RESULTS

Final numerical results of the foregoing analysis are presented in the two tables following. Table 6 is for equilibrium between vapor and solid (ice) and covers the range −160 to 32 F. Table 7 is for equilibrium between vapor and liquid and covers the range 32 F to 212 F. The analysis itself contains all information necessary to compute corresponding tables giving the properties of compressed liquid (or solid) and those of superheated vapor. Space limitations do not permit including such tables, however.

CONCLUSIONS

The authors believe that the results of the present analysis accord with all experimental data used well within their estimated uncertainties. Possibly the results could be improved somewhat by considering other data outside the temperature range of present interest, but we doubt that the improvement would justify the enormous amount of labor involved.

The present formulation involves some revisions to the formulation on which tables presented in a previous paper [2] giving the thermodynamic properties of moist air were based. Such revisions are minor, however, and would not change the numerical values listed in the tables within their estimated uncertainties; they were made chiefly in the interest of consistency.

One of the most bothersome gaps in existing experimental data is the lack of precise knowledge of the relation between the absolute and the international temperature scales. This makes it impossible to state with complete definiteness on what scale the temperatures listed in Tables 6 and 7, for example, belong. For, while the zero-pressure properties were computed for temperatures on

TABLE 7. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION: LIQUID-VAPOR

Temp. Fahr.	Abs. Pressure p_a		Specific Volume, cu ft/lb			Enthalpy, Btu/lb			Entropy, Btu/lb°F			Temp. Fahr.
			Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	Lb/Sq In	In Hg	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
32	0.088586	0.18036	0.01602	3304.6	3304.6	0.00	1075.16	1075.16	0.00000	2.1867	2.1867	32
33	0.092227	0.19778	0.01602	3180.5	3180.5	1.01	1074.59	1075.60	0.00365	2.1811	2.1831	33
34	0.095999	0.19546	0.01602	3061.7	3061.7	2.01	1074.03	1076.04	0.00409	2.1755	2.1796	34
35	0.099908	0.20342	0.01602	2947.8	2947.8	3.02	1073.46	1076.48	0.00812	2.1700	2.1761	35
36	0.10398	0.21166	0.01602	2838.7	2838.7	4.02	1072.90	1076.92	0.00815	2.1644	2.1726	36
37	0.10815	0.22020	0.01602	2734.1	2734.1	5.03	1072.33	1077.36	0.01218	2.1589	2.1691	37
38	0.11249	0.22904	0.01602	2633.8	2633.8	6.03	1071.77	1077.80	0.01620	2.1535	2.1657	38
39	0.11699	0.23819	0.01602	2537.6	2537.6	7.04	1071.20	1078.24	0.01622	2.1480	2.1622	39
40	0.12164	0.24767	0.01602	2445.4	2445.4	8.04	1070.64	1078.68	0.01823	2.1426	2.1588	40
41	0.12646	0.25748	0.01602	2356.9	2356.9	9.05	1070.06	1079.11	0.01884	2.1372	2.1554	41
42	0.13145	0.26763	0.01602	2272.0	2272.0	10.05	1069.50	1079.55	0.02024	2.1318	2.1520	42
43	0.13660	0.27813	0.01602	2190.5	2190.5	11.05	1068.94	1079.99	0.02284	2.1265	2.1487	43
44	0.14194	0.28899	0.01602	2112.3	2112.3	12.06	1068.37	1080.43	0.02483	2.1211	2.1453	44
45	0.14746	0.30023	0.01602	2037.3	2037.3	13.06	1067.81	1080.87	0.02622	2.1158	2.1420	45
46	0.15317	0.31185	0.01602	1965.2	1965.2	14.06	1067.24	1081.30	0.02820	2.1105	2.1387	46
47	0.15907	0.32387	0.01602	1896.0	1896.0	15.06	1066.68	1081.74	0.03018	2.1052	2.1354	47
48	0.16517	0.33629	0.01602	1829.5	1829.5	16.07	1066.11	1082.18	0.03216	2.0999	2.1321	48
49	0.17148	0.34913	0.01602	1765.7	1765.7	17.07	1065.55	1082.62	0.03413	2.0947	2.1288	49
50	0.17799	0.36240	0.01602	1704.3	1704.3	18.07	1064.99	1083.06	0.03610	2.0896	2.1256	50
51	0.18473	0.37611	0.01602	1645.4	1645.4	19.07	1064.42	1083.49	0.03806	2.0844	2.1223	51
52	0.19169	0.39028	0.01602	1588.7	1588.7	20.07	1063.86	1083.93	0.04002	2.0791	2.1191	52
53	0.19888	0.40492	0.01603	1534.3	1534.3	21.07	1063.30	1084.37	0.04197	2.0739	2.1159	53
54	0.20630	0.42005	0.01603	1481.9	1481.9	22.08	1062.73	1084.80	0.04392	2.0688	2.1127	54

Temp. Fahr.	Abs. Pressure p_a		Specific Volume, cu ft/lb			Enthalpy, Btu/lb			Entropy, Btu/lb°F			Temp. Fahr.
			Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	Lb/Sq In	In Hg	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
55	0.21397	0.43564	0.01603	1431.5	1431.5	23.08	1062.16	1085.24	0.04587	2.0637	2.1096	55
56	0.22188	0.45176	0.01603	1383.1	1383.1	24.08	1061.60	1085.68	0.04781	2.0586	2.1064	56
57	0.23006	0.46840	0.01603	1336.5	1336.5	25.08	1061.04	1086.12	0.04975	2.0535	2.1033	57
58	0.23849	0.48556	0.01603	1291.7	1291.7	26.09	1060.47	1086.55	0.05168	2.0485	2.1002	58
59	0.24720	0.50330	0.01603	1248.6	1248.6	27.09	1059.91	1086.99	0.05361	2.0434	2.0970	59
60	0.25619	0.52160	0.01603	1207.1	1207.1	28.09	1059.34	1087.42	0.05553	2.0385	2.0940	60
61	0.26545	0.54047	0.01604	1167.2	1167.2	29.09	1058.78	1087.86	0.05746	2.0334	2.0909	61
62	0.27502	0.55994	0.01604	1128.7	1128.7	30.09	1058.22	1088.30	0.05937	2.0284	2.0878	62
63	0.28488	0.58002	0.01604	1091.7	1091.7	31.09	1057.65	1088.73	0.06129	2.0235	2.0848	63
64	0.29505	0.60073	0.01604	1056.1	1056.1	32.09	1057.09	1089.17	0.06320	2.0186	2.0818	64
65	0.30554	0.62209	0.01604	1021.9	1021.9	33.09	1056.52	1089.60	0.06510	2.0136	2.0787	65
66	0.31636	0.64411	0.01604	988.63	988.65	34.09	1055.97	1090.04	0.06700	2.0087	2.0757	66
67	0.32750	0.66681	0.01605	956.78	956.78	35.07	1055.40	1090.47	0.06890	2.0039	2.0728	67
68	0.33900	0.69021	0.01605	926.06	926.08	36.07	1054.84	1090.91	0.07080	1.9990	2.0698	68
69	0.35084	0.71432	0.01605	896.47	896.49	37.07	1054.27	1091.34	0.07269	1.9941	2.0668	69
70	0.36304	0.73916	0.01605	867.95	867.97	38.07	1053.71	1091.78	0.07458	1.9893	2.0639	70
71	0.37561	0.76476	0.01606	840.45	840.47	39.07	1053.14	1092.21	0.07646	1.9845	2.0610	71
72	0.38856	0.79113	0.01606	813.95	813.97	40.07	1052.58	1092.65	0.07834	1.9797	2.0580	72
73	0.40190	0.81829	0.01606	788.38	788.40	41.07	1052.01	1093.08	0.08022	1.9749	2.0551	73
74	0.41564	0.84626	0.01606	763.73	763.75	42.06	1051.46	1093.52	0.08209	1.9701	2.0522	74
75	0.42979	0.87506	0.01606	739.95	739.97	43.06	1050.89	1093.95	0.08396	1.9654	2.0494	75
76	0.44436	0.90472	0.01606	717.01	717.03	44.06	1050.32	1094.38	0.08582	1.9607	2.0465	76
77	0.45935	0.93524	0.01607	694.88	694.90	45.06	1049.76	1094.82	0.08769	1.9560	2.0437	77
78	0.47478	0.96666	0.01607	673.52	673.54	46.06	1049.19	1095.25	0.08954	1.9513	2.0408	78
79	0.49066	0.99900	0.01607	652.91	652.93	47.06	1048.62	1095.68	0.09140	1.9466	2.0380	79

TABLE 7. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION: LIQUID-VAPOR *(Continued)*

Temp. Fahr.	Abs. Pressure p_s		Specific Volume, cu ft/lb			Enthalpy, Btu/lb			Entropy, Btu/lb°F			Temp. Fahr.
			Sat.Liquid	Evap.	Sat.Vapor	Sat.Liquid	Evap.	Sat.Vapor	Sat.Liquid	Evap.	Sat.Vapor	
t	Lb/Sq In	In Hg	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
80	0.50701	1.0323	0.01607	633.01	633.03	48.06	1048.07	1096.13	0.09285	1.9419	2.0352	80
81	0.52388	1.0665	0.01608	613.80	613.82	49.05	1047.50	1096.55	0.09510	1.9373	2.0324	81
82	0.54118	1.1017	0.01608	595.25	595.27	50.05	1046.93	1096.98	0.09694	1.9328	2.0297	82
83	0.55892	1.1380	0.01608	577.34	577.36	51.05	1046.37	1097.42	0.09878	1.9281	2.0269	83
84	0.57723	1.1752	0.01608	560.04	560.06	52.05	1045.80	1097.85	0.10062	1.9236	2.0242	84
85	0.59604	1.2136	0.01609	543.33	543.35	53.05	1045.23	1098.28	0.10246	1.9190	2.0214	85
86	0.61540	1.2530	0.01609	527.19	527.21	54.04	1044.67	1098.71	0.10429	1.9144	2.0187	86
87	0.63530	1.2935	0.01609	511.60	511.62	55.04	1044.10	1099.14	0.10611	1.9099	2.0160	87
88	0.65575	1.3351	0.01610	496.52	496.54	56.04	1043.54	1099.58	0.10794	1.9054	2.0133	88
89	0.67678	1.3779	0.01610	481.96	481.98	57.04	1042.97	1100.01	0.10976	1.9008	2.0106	89
90	0.69838	1.4219	0.01610	467.90	467.90	58.04	1042.40	1100.44	0.11158	1.8963	2.0079	90
91	0.72059	1.4671	0.01610	454.28	454.28	59.03	1041.84	1100.87	0.11339	1.8919	2.0053	91
92	0.74340	1.5136	0.01611	441.10	441.12	60.03	1041.27	1101.30	0.11520	1.8874	2.0026	92
93	0.76684	1.5613	0.01611	428.36	428.40	61.03	1040.70	1101.73	0.11701	1.8830	2.0000	93
94	0.79091	1.6103	0.01611	416.07	416.09	62.03	1040.13	1102.16	0.11881	1.8786	1.9974	94
95	0.81564	1.6607	0.01612	404.17	404.19	63.03	1039.56	1102.59	0.12061	1.8741	1.9947	95
96	0.84103	1.7124	0.01612	392.67	392.67	64.02	1039.00	1103.02	0.12241	1.8698	1.9922	96
97	0.86711	1.7655	0.01612	381.51	381.53	65.02	1038.43	1103.45	0.12420	1.8654	1.9896	97
98	0.89388	1.8200	0.01612	370.73	370.75	66.02	1037.86	1103.88	0.12600	1.8610	1.9870	98
99	0.92137	1.8759	0.01613	360.30	360.32	67.02	1037.29	1104.31	0.12778	1.8566	1.9844	99
100	0.94959	1.9334	0.01613	350.20	350.22	68.02	1036.72	1104.74	0.12957	1.8523	1.9819	100
101	0.97854	1.9923	0.01614	340.42	340.44	69.01	1036.16	1105.17	0.13135	1.8480	1.9793	101
102	1.0083	2.0529	0.01614	330.96	330.98	70.01	1035.59	1105.59	0.13313	1.8437	1.9768	102
103	1.0388	2.1149	0.01614	321.80	321.82	71.01	1035.01	1106.02	0.13490	1.8394	1.9743	103
104	1.0700	2.1784	0.01614	312.93	312.95	72.01	1034.44	1106.45	0.13667	1.8351	1.9718	104

Temp. Fahr.	Abs. Pressure p_s		Specific Volume, cu ft/lb			Enthalpy, Btu/lb			Entropy, Btu/lb°F			Temp. Fahr.
			Sat.Liquid	Evap.	Sat.Vapor	Sat.Liquid	Evap.	Sat.Vapor	Sat.Liquid	Evap.	Sat.Vapor	
t	Lb/Sq In	In Hg	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
105	1.1021	2.2440	0.01615	304.34	304.36	73.01	1033.97	1106.88	0.13844	1.8309	1.9693	105
106	1.1351	2.3110	0.01615	296.02	296.04	74.01	1033.29	1107.30	0.14021	1.8266	1.9668	106
107	1.1688	2.3798	0.01616	287.96	287.98	75.00	1032.73	1107.73	0.14197	1.8224	1.9644	107
108	1.2035	2.4503	0.01616	280.16	280.16	76.00	1032.16	1108.16	0.14373	1.8182	1.9619	108
109	1.2390	2.5226	0.01616	272.59	272.60	77.00	1031.58	1108.58	0.14549	1.8140	1.9595	109
110	1.2754	2.5968	0.01617	265.24	265.26	78.00	1031.01	1109.01	0.14724	1.8098	1.9570	110
111	1.3128	2.6728	0.01617	258.14	258.16	79.00	1030.44	1109.44	0.14899	1.8056	1.9546	111
112	1.3510	2.7507	0.01617	251.25	251.27	80.00	1029.86	1109.86	0.15074	1.8015	1.9522	112
113	1.3902	2.8306	0.01618	244.57	244.59	80.99	1029.30	1110.29	0.15248	1.7973	1.9498	113
114	1.4305	2.9125	0.01618	238.10	238.12	81.99	1028.72	1110.71	0.15423	1.7932	1.9474	114
115	1.4717	2.9963	0.01618	231.82	231.84	82.99	1028.15	1111.14	0.15596	1.7890	1.9450	115
116	1.5139	3.0823	0.01619	225.73	225.75	83.99	1027.57	1111.56	0.15770	1.7849	1.9426	116
117	1.5571	3.1703	0.01619	219.83	219.85	84.99	1026.99	1111.98	0.15943	1.7809	1.9403	117
118	1.6014	3.2606	0.01620	214.10	214.12	85.99	1026.42	1112.41	0.16116	1.7767	1.9379	118
119	1.6468	3.3530	0.01620	208.54	208.56	86.98	1025.85	1112.83	0.16289	1.7727	1.9356	119
120	1.6933	3.4477	0.01620	203.16	203.18	87.98	1025.28	1113.26	0.16461	1.7687	1.9333	120
121	1.7409	3.5446	0.01621	197.93	197.95	88.98	1024.70	1113.68	0.16634	1.7647	1.9310	121
122	1.7897	3.6439	0.01621	192.85	192.87	89.98	1024.13	1114.10	0.16805	1.7606	1.9286	122
123	1.8396	3.7455	0.01622	187.93	187.95	90.98	1023.54	1114.52	0.16977	1.7566	1.9264	123
124	1.8907	3.8496	0.01622	183.15	183.17	91.98	1022.96	1114.94	0.17149	1.7526	1.9241	124
125	1.9430	3.9561	0.01622	178.51	178.53	92.98	1022.39	1115.37	0.17319	1.7486	1.9218	125
126	1.9966	4.0651	0.01623	174.00	174.02	93.98	1021.81	1115.79	0.17490	1.7446	1.9196	126
127	2.0514	4.1766	0.01623	169.63	169.65	94.97	1021.24	1116.21	0.17660	1.7407	1.9173	127
128	2.1075	4.2910	0.01624	165.38	165.40	95.97	1020.66	1116.63	0.17830	1.7367	1.9150	128
129	2.1649	4.4078	0.01624	161.26	161.28	96.97	1020.09	1117.06	0.18000	1.7328	1.9128	129

TABLE 7. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION: LIQUID-VAPOR *(Continued)*

Temp. Fahr.	Abs. Pressure p_s		Specific Volume, cu ft/lb			Enthalpy, Btu/lb			Entropy, Btu/lb°F			Temp. Fahr.
			Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	Lb/Sq In	In Hg	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
130	2.2237	4.5274	0.01625	137.85	137.87	97.97	1019.50	1117.47	0.18170	1.7289	1.9106	130
131	2.2606	4.6498	0.01625	133.36	133.38	98.97	1018.82	1117.80	0.18330	1.7250	1.9084	131
132	2.3453	4.7750	0.01626	149.38	149.40	99.97	1018.14	1118.11	0.18508	1.7211	1.9062	132
133	2.4061	4.9030	0.01626	145.93	145.93	100.97	1017.79	1118.75	0.18676	1.7172	1.9040	133
134	2.4725	5.0340	0.01626	142.54	142.56	101.97	1017.12	1119.18	0.18845	1.7134	1.9019	134
135	2.5388	5.1679	0.01627	139.19	139.19	102.97	1016.59	1119.56	0.19013	1.7095	1.8996	135
136	2.6066	5.3049	0.01627	135.90	135.92	103.97	1016.01	1119.98	0.19181	1.7056	1.8974	136
137	2.6743	5.4450	0.01628	132.62	132.64	104.97	1015.43	1120.40	0.19348	1.7018	1.8953	137
138	2.7446	5.5881	0.01628	129.06	129.06	105.97	1014.86	1120.82	0.19516	1.6979	1.8931	138
139	2.8165	5.7345	0.01629	125.96	125.96	106.97	1014.24	1121.23	0.19683	1.6942	1.8910	139
140	2.8900	5.8842	0.01629	122.94	122.96	107.96	1013.69	1121.65	0.19850	1.6903	1.8888	140
141	2.9651	6.0371	0.01630	120.01	120.03	108.96	1013.11	1122.07	0.20016	1.6865	1.8867	141
142	3.0419	6.1934	0.01630	117.16	117.18	109.96	1012.52	1122.48	0.20182	1.6828	1.8846	142
143	3.1204	6.3532	0.01631	114.40	114.42	110.96	1011.94	1122.90	0.20349	1.6790	1.8825	143
144	3.2006	6.5164	0.01631	111.70	111.72	111.96	1011.36	1123.31	0.20514	1.6753	1.8804	144
145	3.2825	6.6832	0.01632	109.09	109.11	112.96	1010.77	1123.73	0.20679	1.6715	1.8783	145
146	3.3662	6.8536	0.01632	106.54	106.56	113.96	1010.18	1124.14	0.20845	1.6678	1.8763	146
147	3.4517	7.0277	0.01633	104.06	104.08	114.96	1009.59	1124.55	0.21010	1.6641	1.8742	147
148	3.5390	7.2056	0.01633	101.65	101.67	115.96	1009.01	1124.97	0.21174	1.6604	1.8721	148
149	3.6282	7.3872	0.01634	99.306	99.322	116.96	1008.42	1125.38	0.21339	1.6567	1.8701	149
150	3.7194	7.5727	0.01634	97.032	97.038	117.96	1007.83	1125.79	0.21503	1.6530	1.8680	150
151	3.8124	7.7622	0.01635	94.799	94.815	118.96	1007.24	1126.20	0.21667	1.6493	1.8660	151
152	3.9074	7.9556	0.01635	92.635	92.651	119.96	1006.66	1126.62	0.21830	1.6457	1.8640	152
153	4.0044	8.1532	0.01636	90.529	90.544	120.97	1006.06	1127.03	0.21994	1.6421	1.8620	153
154	4.1035	8.3548	0.01636	88.477	88.493	121.97	1005.47	1127.44	0.22157	1.6384	1.8600	154

Temp. Fahr.	Abs. Pressure p_s		Specific Volume, cu ft/lb			Enthalpy, Btu/lb			Entropy, Btu/lb°F			Temp. Fahr.
			Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	Lb/Sq In	In Hg	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
155	4.2046	8.5607	0.01637	86.480	86.496	122.97	1004.88	1127.85	0.22320	1.6348	1.8580	155
156	4.3078	8.7708	0.01637	84.536	84.552	123.97	1004.29	1128.26	0.22482	1.6312	1.8560	156
157	4.4132	8.9855	0.01638	82.642	82.656	124.97	1003.70	1128.67	0.22645	1.6276	1.8540	157
158	4.5207	9.2042	0.01638	80.798	80.814	125.97	1003.11	1129.08	0.22807	1.6239	1.8520	158
159	4.6304	9.4274	0.01639	79.001	79.017	126.97	1002.51	1129.48	0.22969	1.6204	1.8501	159
160	4.7424	9.6550	0.01639	77.251	77.267	127.97	1001.92	1129.89	0.23130	1.6168	1.8481	160
161	4.8566	9.8869	0.01640	75.548	75.562	128.97	1001.33	1130.30	0.23292	1.6133	1.8462	161
162	4.9732	10.126	0.01640	73.886	73.901	129.97	1000.74	1130.71	0.23453	1.6097	1.8442	162
163	5.0921	10.368	0.01641	72.267	72.283	130.98	1000.13	1131.11	0.23614	1.6062	1.8423	163
164	5.2134	10.615	0.01642	70.690	70.706	131.98	999.54	1131.52	0.23774	1.6027	1.8404	164
165	5.3372	10.867	0.01642	69.153	69.169	132.98	998.94	1131.92	0.23935	1.5990	1.8384	165
166	5.4634	11.124	0.01643	67.654	67.670	133.98	998.35	1132.33	0.24095	1.5956	1.8365	166
167	5.5921	11.386	0.01643	66.194	66.210	134.98	997.75	1132.73	0.24255	1.5920	1.8346	167
168	5.7235	11.653	0.01644	64.770	64.786	135.98	997.16	1133.14	0.24414	1.5887	1.8328	168
169	5.8572	11.925	0.01644	63.382	63.398	136.99	996.55	1133.54	0.24574	1.5853	1.8309	169
170	5.9936	12.203	0.01645	62.029	62.045	137.99	995.96	1133.94	0.24733	1.5817	1.8290	170
171	6.1326	12.487	0.01645	60.710	60.726	138.99	995.36	1134.36	0.24892	1.5782	1.8271	171
172	6.2742	12.775	0.01646	59.425	59.439	139.99	994.76	1134.76	0.25051	1.5748	1.8253	172
173	6.4182	13.070	0.01647	58.168	58.184	141.00	994.15	1135.15	0.25209	1.5713	1.8234	173
174	6.5648	13.370	0.01647	56.944	56.960	142.00	993.56	1135.56	0.25367	1.5679	1.8216	174
175	6.7140	13.676	0.01648	55.750	55.766	143.00	992.96	1135.96	0.25525	1.5644	1.8197	175
176	6.8659	13.987	0.01648	54.586	54.603	144.00	992.36	1136.36	0.25683	1.5611	1.8179	176
177	7.0205	14.305	0.01649	53.450	53.466	145.00	991.75	1136.75	0.25841	1.5577	1.8161	177
178	7.1849	14.629	0.01650	52.341	52.357	146.01	991.14	1137.15	0.25998	1.5543	1.8143	178
179	7.3469	14.959	0.01650	51.260	51.276	147.01	990.54	1137.56	0.26155	1.5509	1.8124	179

TABLE 7. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION: LIQUID-VAPOR *(Concluded)*

Temp. Fahr.	Abs. Pressure p_s		Specific Volume, cu ft/lb			Enthalpy, Btu/lb			Entropy, Btu/lb°F			Temp. Fahr.
			Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	lb/sq in	in Hg	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
180	7.5119	15.295	0.01651	50.803	50.820	148.01	989.93	1137.94	0.84312	1.5475	1.6106	180
181	7.6301	15.537	0.01651	49.173	49.190	149.02	989.38	1138.34	0.84668	1.5442	1.6089	181
182	7.6514	15.904	0.01652	49.146	49.165	150.02	988.72	1138.74	0.84825	1.5408	1.6071	182
183	8.0256	16.341	0.01652	47.197	47.204	151.02	988.18	1139.14	0.85781	1.5375	1.6053	183
184	8.2035	16.703	0.01653	46.229	46.246	152.03	987.60	1139.53	0.86937	1.5341	1.6035	184
185	8.3845	17.071	0.01654	45.294	45.311	153.03	986.99	1139.92	0.87093	1.5308	1.6017	185
186	8.5688	17.446	0.01654	44.361	44.396	154.04	986.50	1140.32	0.87248	1.5275	1.6000	186
187	8.7565	17.828	0.01655	43.469	43.506	155.04	985.87	1140.71	0.87404	1.5242	1.7982	187
188	8.9476	18.218	0.01656	42.619	42.636	156.04	985.07	1141.11	0.87559	1.5209	1.7965	188
189	9.1422	18.614	0.01656	41.769	41.786	157.05	984.45	1141.50	0.87713	1.5176	1.7947	189
190	9.3403	19.017	0.01657	40.939	40.956	158.06	983.84	1141.89	0.87868	1.5143	1.7930	190
191	9.5420	19.429	0.01658	40.128	40.145	159.06	983.22	1142.28	0.88022	1.5111	1.7913	191
192	9.7473	19.848	0.01658	39.337	39.354	160.06	982.61	1142.67	0.88176	1.5078	1.7896	192
193	9.9563	20.271	0.01659	38.563	38.580	161.06	982.00	1143.06	0.88330	1.5045	1.7878	193
194	10.169	20.704	0.01659	37.807	37.824	162.07	981.38	1143.45	0.88484	1.5013	1.7861	194
195	10.386	21.145	0.01660	37.069	37.086	163.08	980.76	1143.84	0.88638	1.4980	1.7844	195
196	10.606	21.594	0.01661	36.348	36.365	164.08	980.15	1144.23	0.88791	1.4949	1.7828	196
197	10.830	22.050	0.01661	35.643	35.660	165.08	979.54	1144.62	0.88944	1.4917	1.7811	197
198	11.058	22.515	0.01662	34.954	34.971	166.09	978.91	1145.00	0.89097	1.4884	1.7794	198
199	11.290	22.987	0.01663	34.281	34.298	167.10	978.29	1145.39	0.89250	1.4852	1.7777	199
200	11.526	23.468	0.01663	33.623	33.640	168.10	977.68	1145.78	0.89402	1.4820	1.7760	200
201	11.767	23.957	0.01664	32.980	32.997	169.11	977.05	1146.16	0.89554	1.4788	1.7744	201
202	12.011	24.455	0.01665	32.351	32.368	170.11	976.43	1146.54	0.89706	1.4756	1.7727	202
203	12.260	24.961	0.01665	31.737	31.754	171.12	975.81	1146.93	0.89858	1.4725	1.7711	203
204	12.513	25.476	0.01666	31.136	31.153	172.12	975.19	1147.31	0.90010	1.4693	1.7694	204

Temp. Fahr.	Abs. Pressure p_s		Specific Volume, cu ft/lb			Enthalpy, Btu/lb			Entropy, Btu/lb°F			Temp. Fahr.
			Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	lb/sq in	in Hg	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
205	12.770	26.000	0.01667	30.549	30.566	173.13	974.56	1147.69	0.90161	1.4662	1.7678	205
206	13.031	26.532	0.01667	29.974	29.991	174.14	973.94	1148.08	0.90312	1.4631	1.7662	206
207	13.297	27.074	0.01668	29.413	29.430	175.14	973.32	1148.46	0.90463	1.4600	1.7646	207
208	13.568	27.625	0.01669	28.863	28.880	176.15	972.69	1148.84	0.90614	1.4568	1.7629	208
209	13.843	28.185	0.01669	28.326	28.343	177.16	972.06	1149.22	0.90765	1.4536	1.7613	209
210	14.123	28.754	0.01670	27.801	27.818	178.17	971.43	1149.60	0.90915	1.4506	1.7597	210
211	14.407	29.335	0.01671	27.287	27.304	179.17	970.81	1149.98	0.91065	1.4474	1.7581	211
212	14.696	29.921	0.01671	26.784	26.801	180.18	970.17	1150.35	0.91215	1.4444	1.7565	212

the absolute scale, they were used in conjunction with calorimetric data for temperatures on the international scale. Still, in many cases integration with respect to absolute temperature, or to the logarithm of absolute temperature, of an integrand whose values do not depend strongly on temperature was involved. In such cases the results are to be regarded as valid for temperatures on the absolute scale, even though the integrands were evaluated for temperatures on the international scale. This is true in the case of our calculated vapor pressures. It is hoped that the relation between the two scales will ultimately be defined more precisely by further experimentation. A necessary preliminary seems to be to improve the specifications for the international scale to insure a greater degree of reproducibility.

PHYSICAL CONSTANTS AND CONVERSION FACTORS

It seems appropriate to collect the numerical values of the various physical constants and conversion factors used in the preceding analysis.

Pressure-volume product, pv (0°C, 0 atm):	$RT_o = 22414.6$ cm³atm/mole
Ice-point temperature:	$T_o = 273.16$°K
Standard gravitational acceleration:	$g_o = 980.665$ cm/sec²
Density of Mercury (0°C, 1 atm):	$D_o = 13.59504$ g*/cm³
Molecular weight of water:	$M_w = 18.0160$ g/mole
Second Radiation Constant:	$hc/k = 1.43848$ cm °K

1.033223 kg/cm²atm
860 kcal/int (mean) kwhr
1.00020 abs kwhr/int (mean) kwhr
1 Btu kmol°K/kcal mol°/R

Universal Gas Constant: $R = 1.98581$ Btu/mol°R

2.54 cm/in
453.5924 g/lb
1.8°R/°K
778.18 ftlb/Btu
1.000026 int (certified NBS) kwhr/int (mean) kwhr

ACKNOWLEDGMENT

The present analysis was a necessary step in the formulation of the thermodynamic properties of moist air which was undertaken as a cooperative investigation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The support and encouragement of the Society are gratefully acknowledged. The paper itself is submitted as a contribution from the University of Pennsylvania Thermodynamics Research Laboratory, and for the active support and encouragement of the Navy Department, Bureau of Ships, which have made the establishment of this laboratory possible it is a pleasure to acknowledge appreciation.

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DISCUSSION

W. N. MURRAY,[2] Moline, Ill: I would like to know whether from any of your work you obtained new values for the ratio of c_p to c_v. The values in the literature are not sufficiently accurate down to low pressures and I was wondering whether that could have been obtained out of any of your measurements.

DEAN GOFF: We did not compute the ratio of c_p to c_v but, in addition to tables giving values of the various thermodynamic properties at saturation, we have presented a formulation from which the properties of the superheated vapor, including the ratio c_p/c_v, can be readily computed.

MR. MURRAY: It would be very useful.

DEAN GOFF: I shall take advantage of Mr. Murray's question to emphasize one feature of our formulation which we regard as having particular importance. This

[2] The Herman Nelson Corp.

is the extent to which available spectroscopic data can be correlated with the best calorimetric data such as those obtained from the painstaking research at the Bureau of Standards. This enhances our confidence in the general procedure of relying on spectroscopic data for zero-pressure properties and then using calorimetric data to determine the effect of pressure.

Since my paper was presented a paper has been published[3] in which the author accepts Beattie's findings regarding the relation between the absolute and the international temperature scales and uses it to transfer measured saturation pressures to the absolute scale before attempting to correlate them with other data through the identical relations of thermodynamics. Because of Mr. Keyes' familiarity with these other data, the manner in which they were obtained, and the way in which they should be interpreted, it may be that his formulation of the saturation pressure versus temperature relation should be regarded as superior to ours, at any rate in the range 32 to 212 F, Keyes used available saturation pressure data as a principal support for his formulation; we have used those in the range 0 to 125 C principally to test the reliability of our overall formulation, which in other respects must be regarded as superior to his. The differences between the two formulations are extremely small.

[3] Thermodynamic Properties of Water Substance 0 to 150 C, by F. G. Keyes. (*The Journal of Chemical Physics*, Vol. 15, No. 8, August, 1947.)

1287

No. 1287

THERAPEUTIC USES OF LOW TEMPERATURE

By FREDERICK M. ALLEN, M.D., NEW YORK, N. Y.

BY FAR the leading causes of death in the civilized world are the disorders of metabolism, which include diabetes and high blood pressure, both of them causes of arteriosclerosis. Although the author's field of work is this group of metabolic diseases and not surgery, a considerable number of deaths are due to arteriosclerotic gangrene of the feet, and experiments which were originally aimed at reducing the mortality from this cause have kept him occupied for a number of years with surgical problems of wounds and shock. But since metabolism comprises the total chemical processes of life, a wide range of studies concerning modifications of tissue metabolism by temperature still fall within the author's specialty.

If treatment of the subject begins fundamentally with the biology of temperature, it is large enough to occupy several volumes. Elevation or depression of temperature can harm or sometimes terminate life processes, but in general there is a contrast between the two directions. The upward range of endurable temperature is slight from the standpoint of a physicist. A few thermophile organisms such as some bacteria can survive temperatures close to boiling, but not far from that point all life is killed positively and beyond recovery. But on the downward range the physicist may be invited to do his worst; there is no universal killing by cold even at absolute zero; the stopping of life processes is reversible, or when death does occur it is often due to complicating factors.

One important complicating factor is hydration, conducing to the mechanical tearing of protoplasm by ice crystal formation. Therefore, organisms which are easily killed by cold when wet may be highly resistant when dry. Another variable is the capacity of protoplasm for supercooling, whereby not only low organisms but also the human skin may withstand temperatures 10 to 15 deg below the true freezing point without freezing. Another peculiar physical alteration is represented in the killing of some organisms at a few degrees below freezing and their survival if they are reduced rapidly to the tempera-

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ture of liquid gases, because there viscosity or other protoplasmic changes prevent freezing. In general, resistance diminishes with ascent in the biological scale, but variations among species are enormous and unexplained. Among arthropods, contrary to popular belief, flies, ants, bees and all common insects are killed by very slight freezing, but tardigrades have phenomenal ability to survive the temperature of liquid hydrogen in either a wet or a dry condition. Reptiles, amphibia and fish also die with ordinary freezing. A popular error arises from the fact that a frog or fish may survive after being temporarily caught in ice, but if the body is frozen through, the heart is stopped and does not resume. Some tropical varieties are killed by chilling far above freezing, and the functional response to temperature varies extremely among different species. The metabolism of cold-blooded animals is typically accelerated by heat and retarded by cold, as evidenced by the active or sluggish behavior under the respective conditions. But fish such as the trout and salmon are intensely active in icy water which would halt muscular, nervous and glandular functions in most species. Since these fish also are killed by freezing, their thermal margin between optimal life conditions and death must be peculiarly narrow. Some micro-organisms but no higher species are able to carry on life processes at temperatures appreciably below zero Centigrade, the lowest recorded limit being the growth of a fungus at −10 C (+14 F). Partial life functions in the form of enzyme activity can continue at various low temperatures not only in primitive but also in higher species, a familiar example being the *tenderizing* of mammalian muscle in prolonged refrigeration. Still another variable remains to be considered, namely, time, since very commonly a temperature which is endured for a short time becomes fatal in a longer time. This time factor appears strange if very low temperatures actually abolish all protoplasmic functions; but one of several hypotheses is that death results from the continuing activity of some enzymes or accumulation of their products.

The summary of everything is that while intense heat always kills, intense cold may either destroy or preserve life. The frigid zone of biology has fantastic possibilities. Bacteria are said to have been cultivated from inside the trunks of mammoths which had been frozen in the Siberian ice for thousands of years. There can be no positive denial of the speculation of Arrhenius that the primordial germs of life might have been carried to this planet from outer space through eons of time at the lowest limits of cold. But the topic requires a return to a concrete question: what is the relation of these diverse facts to practical medicine and the physiologic and pathologic processes with which it deals?

The effects of low temperature upon the highest species, including man, must be studied according to a threefold classification, namely, the effects upon the whole organism, upon local parts, and upon isolated parts.

EFFECTS OF COLD UPON ENTIRE ORGANISM

For all homoiotherms or warm-blooded animals, there is an absolute low limit of temperature compatible with life. This ranges from about 13 C (55.4 F) in the rat to 22 C (71.6 F) in the dog. When the lethal temperature is reached even momentarily, there is a sudden stoppage of respiration, then

of the heart, and there is no recovery upon warming. Hibernators retain the exceptional power of reverting practically to a cold-blooded behavior, with the characteristic torpidity on cooling, and some of them can withstand temperatures slightly below zero Centigrade, but freezing of vital organs is fatal. In all species the time factor is very important, all temperatures above the lethal point being tolerated for various times. There is roughly an inverse relation between time and temperature; the lower the temperature the more quickly it is fatal.

Any marked reduction of the body temperature of a warm-blooded animal is injurious and ultimately fatal. Therefore, defensive reflex mechanisms are active in combating any reduction. These include especially diminution of heat loss by constriction of superficial blood vessels and increase of heat production by the involuntary muscular activity of shivering. Dr. Temple Fay has reduced the human temperature to the lowest known limits, namely, to 24 C (75.2 F), which must be close to the lethal point. He was the bold pioneer discoverer of the torpid, almost unconscious state of *artificial hibernation*, which differs in important respects from true hibernation, and he can tell more about bodily hypothermia than anybody else. Only one central fact need be considered here. The utmost voluntary and involuntary efforts are justified to preserve body temperature, because of the ultimate fatal effect of reduction, but paradoxically these efforts may in themselves be fatal. Particularly by using sedatives to dull the reflexes and allow the body temperature to fall with a minimum of defensive reactions, Dr. Fay has been able safely to maintain rectal temperatures of 85 F or even lower for several days or a week, whereas a person exposed under ordinary conditions would die within a shorter time from exhaustion due to voluntary and involuntary efforts to prevent such a fall of temperature.

EFFECTS OF COLD UPON LOCAL PARTS

Cold applied to a local area causes first a pallor due to vasoconstriction, with a tendency to reflex contraction of superficial vessels throughout the body. Later the local capillaries dilate, causing redness, though the larger vessels remain contracted. Sufficient intensity or duration of cold causes local edema, or wheal, and the *flare* described by Lewis is a surrounding redness due to capillaries dilated by formation of histamine-like substances in the tissues. Results are determined by the usual three factors of temperature, time and complications. Freezing with its crystal formation is particularly serious but by no means necessarily fatal, since a cat's leg has been experimentally frozen so hard that it would ring when struck and yet could recover with only brief inflammation and no permanent damage; and the similar survival of human fingers and ears after brief freezing is well known. The lowest temperatures kill tissues in the shortest time, but many persons will be surprised that such sensitive structures as liver and brain tissues and major blood vessels have been frozen for one minute with solid carbon dioxide without permanent damage. Important complicating factors are wetness, which when long continued can cause tissue changes even without severe temperature; also dependent position, obstructed circulation and nutritive deficiencies. Thus *immersion foot*

occurs particularly in soldiers or sailors whose feet have been wet, cold, and in a cramped dependent position for several days.

After all these forms of injury, the first result of warming is a rush of blood. Then, in the severest cases, the arteries clot shut and necrosis of tissues from lack of blood is seen in the form of ulcers or gangrene. In less severe cases chronic nervous and circulatory disturbances persist for months, manifested by anesthesia, hyperesthesia, red, white or bluish mottled colors, and sometimes pain to a degree requiring amputation. Minor persistent lesions are familiar in the form of chilblains. After the mildest injuries, complete healing occurs more or less rapidly.

EFFECTS OF COLD UPON ISOLATED TISSUES

No formal distinction of this kind has heretofore been made in the literature, but there is a very sharp difference in behavior of tissues when they are removed from the nervous and circulatory control of the body. Evidently much of the local injury from cold represents a breakdown of defensive reactions. Not only are nervous and circulatory disturbances impossible but also the chemical response seems to be lacking, since tissues chilled outside the body do not excite the characteristic wheal and flare when reimplanted in the body. Also the tolerance of severe and prolonged freezing, such as would be unthinkable for any part connected with the body, proves that mammalism cells have by no means lost the resisting power inherited from primitive organisms. Thus cancer cells have survived freezing for two years. Human spermatozoa have regained activity after freezing at the nearest possible approach to absolute zero. More complex structures also reproduce the primitive behavior. Desiccation, which is considered most inimical to parts of the intact body, assists in the preservation of skin grafts for days or weeks at the temperature of solid carbon dioxide (Webster). The sensitiveness of blood vessels appears to be responsible for the necrotic effects of freezing, but large vein segments for surgical use are preserved for months at the same low temperature and do not then cause clotting of blood (Blakemore and Lord).

Much work will be required to determine what structures can be preserved, at what temperatures and for how long. In some instances, as mentioned, there is the paradox that temperatures near freezing may be more injurious than those far below freezing, perhaps because of irregular disturbances of cell enzymes. Limbs can be separated from their nervous and circulatory connections by a tight tourniquet, without amputation. Thus they will not display the defensive reactions of limbs which retain connections. Severe arteriosclerosis or wounds appear to make a partial change of this kind. It is not feasible to freeze or desiccate such limbs, but at least within a range from about 5 C (41 F) upward there is a reversal of the usual law. That is, tissues connected with the body are killed by the lowest temperature in the shortest time, but isolated tissues or limbs within the range mentioned survive longer in proportion as the temperature is lower. This survival can be very considerable, as the author was able to preserve ligated animals' legs in ice water for 54 hr, and Brooks and Duncan kept rats' tails for 96 hr. It is

uncertain how much longer the survival might be possible if technical difficulties could be overcome.

THERAPEUTIC USES

Considering these effects not as physiological curiosities but as powerful therapeutic tools, advance can be made far beyond the previous trivial uses of cold. The benefits of reduction of general body temperature should be left to Dr. Temple Fay for the most authoritative discussion. The author's impression is that the greatest usefulness exists in the treatment of shock, where the former tradition of warming patients has been reversed and a moderate reduction of both environmental and rectal temperatures is endorsed by all of the most recent investigators.

The author's own field of experience, namely local refrigeration, comprises far lower temperatures than can be applied to the body as a whole. The effects are correspondingly more powerful, within the local limitations. Obviously the chilling can be most radical and penetrating when blood flow is stopped by a tourniquet, but even without this aid experiments have proved that cold applied to the skin can produce considerable reductions of deep temperature, mostly by direct conduction but partly by reflex contraction of blood vessels. Refrigeration, halting all metabolism, places the tissues in a state of suspended animation, temporarily as reactionless as if they were dead. Thus it has the unique property of simultaneously inhibiting pain, shock, exudation, infection, necrosis, and formation and absorption of toxins. Hence it is used in a growing list of conditions which need one or all of these effects, as follows:

1. *Surgical anesthesia:* Cold is the only known anesthetic which anesthetizes not only nerves but also the entire protoplasm, and it thus provides the only shockless surgery. It was first used for this purpose, which was the origin of the research, namely amputation in dangerous gangrene cases, particularly those which were formerly given up as *inoperable* because of weakness and shock. The method consists essentially in applying a tight tourniquet of narrow rubber tubing well above the proposed level of amputation, the skin having been previously chilled to prevent pain from the compression. As soon as the tourniquet has been fastened, the entire limb is buried in ice to a couple of inches above the tourniquet. Any preexisting pain, even the most excruciating, ceases, and the patient is comfortable during the refrigeration period, which is the time required for cold to penetrate and deaden the deepest nerves. This time may range from one hour for an ankle to three hours for the thickest thigh. The ice is removed in the operating room and the amputation performed without pain or shock, the tourniquet being released just before final closure of the wound. Moderate cooling of the stump to a skin temperature of about 15 C or 59 F is continued for a day or two, to control pain, inflammation and possible infection or necrosis. Many surgeons have agreed that the mortality in poor-risk amputation cases has been greatly reduced by this method (in the City Hospital, N. Y., a change from about 80 per cent deaths to about 80 per cent recoveries).

2. *Infection and intoxication:* As an example a patient may enter the hospital nearly dead with infected gangrene. The leg is packed in ice. The intoxication is halted so that within 24 or 48 hr fever, delirium or unconsciousness are cleared up and the patient is eating. A tourniquet can be applied for still more absolute blocking of toxic absorption, and the amputation may be deferred for days or even weeks if necessary. Recently this method has been tried for other dangerous infections, especially gas gangrene, with encouraging results.

3. *Brief preservation after wounds:* As an example, a man may arrive in the hospital in a dying state after amputation of both legs by a railway train. He revives under transfusions and other treatments, and pain is relieved by packing the stumps in ice. He is in condition for successful surgery 58 hours later, the wounds being fresh and free from odor. (Case of Dr. R. T. McElvenny).

4. *Prolonged preservation after wounds:* For example, if men sustain fractures of the pelvis by being crushed by trucks or falling from heights. In each instance the main artery of one leg is damaged and clotted shut. Amputation would be fatal in the presence of other severe injuries and shock. In cases reported, packing in ice prevented gangrene of the leg for 2 or 2½ weeks while other injuries were being treated and strength built up; then amputation was followed by recovery. (Cases of Dr. H. E. Mock.) A seaman on a destroyer had his femoral artery and vein severed in the upper thigh by a shell fragment. Refrigeration kept him comfortable and free from gangrene until amputation became possible after one month. (Case of naval surgeons Ottaway and Foote.)

5. *Saving of limb:* A blast accident tore off one of a soldier's hands, mangled the other and inflicted body injuries. A quick wrist amputation might not have been fatal, but both forearms were packed in ice. After 2½ weeks there was sufficient recovery of strength for reconstructive surgery. The lacerated tissues were as fresh as at the time of the wound. One clean amputation was completed and the other hand was elaborately repaired so as to restore nearly normal function, thus avoiding *the horror of no prehensil limb.* (Case of Dr. I. M. Nachlas.) Experimentally, it has already been proved possible to amputate a dog's leg, keep it in a refrigerator for 24 hr, and then restore it in position and usefulness. Future surgery may therefore hope to replace severed limbs, perhaps some time after the accident.

6. *Embolism:* Embolism in a limb means that an artery is stopped by a clot or other solid object carried to it from elsewhere. Thus it occurs in heart disease and other conditions where the patient may be unfit for immediate operation. Refrigeration has preserved such limbs from gangrene for 2 to 6 weeks, until conditions for amputation were more favorable. (Cases of Dr. L. W. Crossman.)

7. *Shock:* In shock arising from crushing or other severe injuries of limbs, life may be saved by refrigeration, as proved definitely by animal experiments. This is part of the routine treatment of the *crush injury* which became familiar during bombing raids in England. Pressure by means of casts or tight bandages is commonly used for this same purpose, but though it is helpful for shock it does not control pain and may tend to increase tissue necrosis and infection. Cold can be used either alone or with pressure to obtain the combined effects.

8. *Injuries due to cold:* Frozen areas or "immersion foot" should not be warmed too fast or too far. Increase of pain, swelling, thrombosis and necrosis is the result. In the present war, Webster and other Canadian and English investigators learned that by keeping the affected parts moderately cool, with skin temperatures around 60 to 70 F, sometimes for several weeks, the neurovascular symptoms are greatly relieved and many amputations prevented. In the severest cases, a recent therapeutic advance is based on the previously mentioned fact that tissues are not directly killed by cold but die as a secondary result of thrombosis of vessels. The new treatment (Lange, Loewe and others) consists in preventing blood clotting by administering heparin for several days. The result is complete healing of frozen parts such as heretofore have become hopelessly gangrenous. But often it will be necessary to thaw the frozen parts at only a few degrees above freezing in order to prevent both thrombosis and necrosis in the interval before heparin can be obtained. Also the above-described need for cooling in the later stages is not altered.

9. *Burns:* Burns preeminently create a need for the unique potency of cold for controlling pain, shock, infection, necrosis and intoxication. Anybody can readily convince himself that the best emergency treatment is to plunge a burned part into

the coldest obtainable water or apply ice. There is not only the best immediate relief of pain and inflammation but also the best influence on the subsequent course. While any other desired treatments may be combined with this, the later management includes moderate cooling for a number of days or weeks, somewhat as in frostbite cases. A recent case report by Dr. I. Kross * is part of the basis of hope that this method will revolutionize treatment and results in severe burns.

10. *Military surgery:* Obviously these enumerated types of injuries occur in war to the extent of thousands of cases. The weapons now in use have made limb wounds and burns particularly numerous. The examples from civilian surgery illustrate the value of refrigeration for painless and shockless transportation of wounded and for a wide variety of operations and treatments. Beyond question many lives and limbs have been lost as a result of the failure of the official scientific advisory committees to recommend methods and apparatus for this purpose. The arbitrary objection that refrigeration is not feasible near the fighting fronts has been answered by the refrigeration of food and drinks in advanced areas, the rapid transportation of wounded to equipped hospitals, and the actual use of the method on the initiative of medical officers with improvised apparatus. Tropical climate accentuates the need. Forthcoming reports will show the benefits thus derived from surgical refrigeration during the recent Pacific campaigns.

The preceding descriptions have been based mostly on the use of ice, which is the most primitive and still the most common method. Special apparatus is a convenience and in some respects almost a necessity for both civil and military treatments. Doubtless in this field there is more scientific interest than financial gain, and the machines of a manufacturer in Buffalo, N. Y., are only ones now available. They circulate an alcohol-water mixture at selected temperatures through blankets, metal applicators or air chambers. Any such apparatus must be thoroughly constant and dependable in temperature, the automatic reliability being not only a therapeutic necessity but also an important saver of nursing work. Other desirable features are light weight, noiselessness, adaptability and minimum interference with a patient's comfort and movements. All these qualities are put to test in treatments requiring weeks or months of regulated temperature, for which ice bags are extremely inefficient and laborious. The existing apparatus has made possible the results obtained by the author in the City Hospital (New York), and apparently no more than minor improvements will be needed unless some radically new engineering principle comes into view. The author desires chiefly to leave the general conception of refrigeration as a scarcely explored borderland of suspended or modified vital processes, and to suggest the scientific and humanitarian benefits that can accrue if anybody in the industry concerned with production and control of low temperatures chooses to finance further investigations in this field.

DISCUSSION

C. P. YAGLOU, Boston, Mass. (WRITTEN): Dr. Allen has brought into our Society first hand information on new possibilities of refrigeration in the therapeutic field. Although local body cooling by means of ice bags has long been in use for the relief

* *Journal American Medical Association*, May 5, 1945, p. 19.

of pain and swellings, nothing exciting developed until just before and during the war, when anything that looked at all promising in saving life or limb was given a fair trial.

As Dr. Allen implies in his most informative paper, some of the results in human refrigeration have far exceeded expectations, while others are still in the speculative state. Old conceptions about cold and disease, limits of adaptability to cold effects of sudden temperature changes, etc., will have to be revised in the light of recent findings. A fruitful field has been opened for investigation and the ultimate findings will undoubtedly affect the practice of air conditioning for comfort, health and therapeutics. I would therefore propose that Dr. Allen's request for assistance and cooperation be given cordial consideration by our Society.

1288

No. 1288

HEATING AND VENTILATING FOR TRANSPORT AIRPLANES

By B. M. Brod,* New York, N. Y.

Design Conditions

IN THE selection of outside design temperatures for an aircraft heating system, most considerations of local conditions are disregarded in favor of the more general climatic conditions and the altitude at which the aircraft is to fly. For example, certain Flagships operating throughout the United States and into Canada and Mexico have a service altitude of about 15,000 ft maximum. The heating system is sized to provide a comfortable interior when flying through outside air at −40 F. Operation at 20,000 ft maximum is anticipated in the near future. The outside design temperature will be −60 F.

The design inside condition can be found fairly accurately from the A.S.H.V.E. Comfort Chart. Although it is usually considered to be +70 F, we cannot overlook the fact that, with the inside relative humidities presently encountered, a warmer temperature is more desirable. Therefore, in the selection of the interior design condition, inside relative humidities must be estimated before a truly satisfactory standard can be set up. Means of increasing humidities will be discussed later.

Preparing the Heat Load

Unfortunately, customary practice today in the sizing of an airplane's system is still too rough and ready. Although seldom undersized, systems are frequently oversized, resulting in poor control and uncomfortably high heater discharge temperatures. A rather accurate method does exist and has been suggested.[1]

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[1] Calculation of the Heat Loss from an Airplane, by Wilbur W. Reaser, Douglas Aircraft Co., presented before the A.S.M.E. on June 18, 1941, Kansas City, Mo.

Presented at the 52nd Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January 1946.

Fig. 1 shows a typical section of airplane structure, the heat flow through which is illustrated in Fig. 2, being likened to electrical flow for purposes of representation. This flow can be determined empirically from the standard heat flow formula:

$$\frac{1}{U} = \frac{1}{f_1} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \cdots + \frac{x_n}{k_n} + \frac{1}{f_o} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

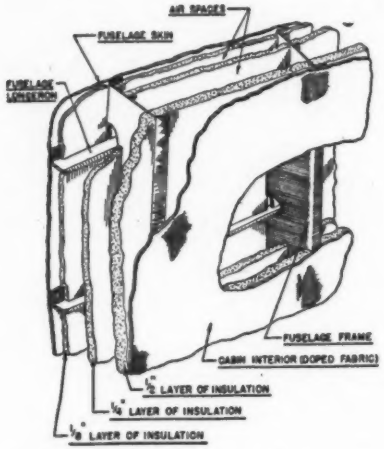


FIG. 1. TYPICAL AIRPLANE STRUCTURE AND INSULATION

where

U = Overall (air to air) coefficient of heat transmission Btu per (hr) (sq ft) (deg temp. difference)

f_1 = Inside (air to wall) film conductance Btu per (hr) (sq ft) (deg temp. difference)

f_o = Outside (wall to air) film conductance Btu per (hr) (sq ft) (deg temp. difference)

$x_1 x_2$, etc. = Thickness of any given wall material

k = Thermal conductivity Btu per (hr) (sq ft) (deg temp. difference per inch)

At first glance it would appear that the heat flow through the frames and longerons could be neglected since the thickness, for example, of the frame is usually in the nature of 0.040 in. spaced about 20 in. apart. Further examination shows, however, that an error of up to 400 per cent may result from such treatment, since the value of k for aluminum is 813 Btu per (hr) (sq ft) (Fahrenheit deg per inch), whereas for air space it is only 1.1 Btu. Thus parallel flow calculations must be made for all paths of heat flow in order to arrive at an accurate heat loss. Since the speed of the airplane is so

great, an almost negligible surface coefficient exists for the outside of the airplane and, hence, the outside surface coefficient can be neglected.

The problem is further complicated by the lack of accurate or reliable figures for the thermal coefficients of many of the materials used for sound-proofing; and by the different shape, cross-sectional area, lengths, widths, etc., of aircraft structure. At present, considerable effort and time are required to make an accurate estimate of the heat loss, but since malfunctioning of the system will result from incorrect sizing, expenditure of the time and effort are well worth while.

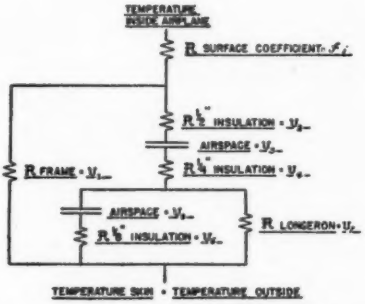


FIG. 2. HEAT FLOW THROUGH TYPICAL AIRPLANE STRUCTURE

VENTILATION REQUIREMENTS

Even after the heat loss from the airplane is known, the size of the heating plant cannot be determined until the quantity of ventilating air is computed. This is a matter which requires considerable thought since, to prevent objectionable odors, tobacco smoke, etc., as much air as possible is desired, whereas for economy of operation, reduction of humidity, etc., the minimum amount is preferable.

The humidity problem has become increasingly important of late and is likely to become even more important in the near future. Since each person gives off approximately 0.153 to 0.492 lb of moisture per hour at room temperature, the occupants of the cabin are an excellent source of humidity. The relative humidity which can be maintained in the cabin at 70 F (neglecting the problems of condensation on walls and windows) can be predicted from the following formulas:

$$\Phi = 100 \frac{p_w}{p_s} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (2)$$

$$p_w = p_s \left(\frac{P_c}{P_o}\right) + \frac{1.45}{n} \quad \ldots \quad (3)^*$$

where

Φ = Relative humidity in cabin, per cent
p_w = Vapor pressure of moisture in cabin, inches mercury
p_s = Vapor pressure of saturated steam at cabin temp., inches mercury
p_{s_o} = Vapor pressure of saturated steam in outside air temp., inches mercury
P_c = Cabin pressure, inches mercury
P_o = Outside pressure, inches mercury
n = Ventilation rate, cfm per person

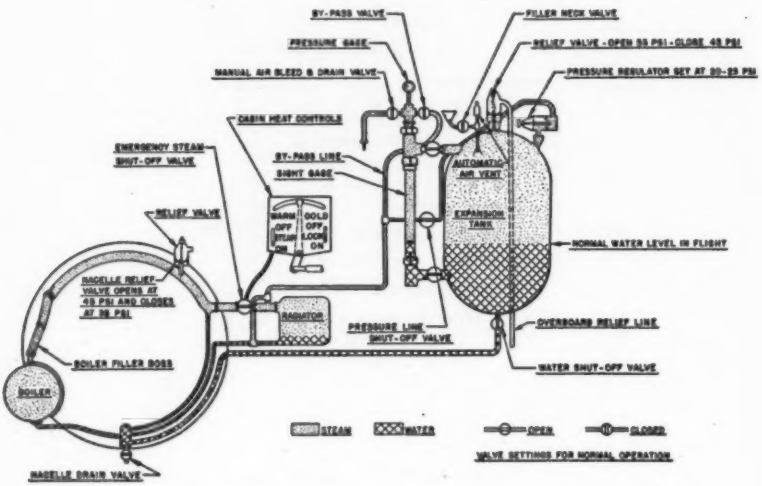


FIG. 3. DIAGRAMMATIC LAYOUT OF AIRPLANE STEAM HEATING SYSTEM

From Equation 3 it is evident that the lower the ventilation rate per occupant, the greater will be the relative humidity per person. Actually, with an airplane flying at 20,000 ft true altitude through saturated air, with the cabin maintained at a pressure equivalent to 8,000 ft altitude and with a ventilation rate of 3 cfm of outside air per person, the relative humidity in the cabin would be approximately 65 per cent.

Obviously such a low ventilation rate in an area with such a high occupancy would soon be made intolerable by smoke and body odors, and hence a higher value must be used. At present, ventilation rates as high as 50 cfm per person are used, which accounts for the freshness of the air and the lack of smoke haze, as well as the excessive dryness in present day airplane cabins.

* This formula true only for 70 deg cabin.

The low humidity in aircraft cabins today, while quite satisfactory for low-level flights in the summer time, presents a very serious problem for long range, high altitude flights throughout the year. Trans-Atlantic flight crews often refer to what they term dry-eyeballitus as a contributing factor in crew fatigue, and the American Airlines Medical Department has on a statistical basis named the excessive dryness of the air on trans-Atlantic flights as a

FIG. 4. AMERICAN AIRLINES TYPE AIRCRAFT STEAM BOILER

cause for the large number of temporary minor respiratory ailments such as colds and drying of the mucous membranes.

In the operation of aircraft for long range flights, therefore, a means will have to be found for increasing the relative humidity in the cabins. Recirculation of the air constitutes one way of increasing the relative humidity, but since a considerable amount of apparatus is required to purify the reused air, it may prove advisable rather to use all outside air and to provide an additional source of moisture. This is one of the problems which the industry is facing today and one for which a final solution has not as yet been worked out.

On the basis of ventilation figures used for night clubs, buses, trains, and similarly highly occupied spaces, it is apparent that at least 20 cfm of equivalent outside air must be supplied per cabin occupant in order to maintain

the desired cabin atmosphere. It appears today that the most economical arrangement is to provide all outside air and to provide additional moisture to the air when required. Development of lightweight air purifiers, capable of removing odors and smoke haze, may be extremely useful for air transport use.

Another limiting factor in the equivalent ventilation rate is the heater discharge air temperature. For reasons of safety, a figure of 350 F is usually used as the maximum temperature in an aluminum alloy duct, and a figure of below 300 F for most synthetic rubber ducts. The minimum air quantity which will result in a safe discharge air temperature can be determined from the following familiar formulas when the cabin heat loss is known:

$$W = \frac{H}{c_p \Delta t} \quad \ldots \quad (4)$$

$$Q_m = \frac{W}{60 v_a} \quad \ldots \quad (5)$$

where

W = Air flow, pounds per hour
H = Total heat loss from airplane, Btu/hr
c_p = Specific heat of air, Btu/lb
Δt = Temperature difference between inside and outside air
Q_m = Air flow, cubic feet per minute
v_a = Specific volume of air, cubic feet per pound

Since the weight of air supplied does not vary with pressure, density, or temperature, the condition under which v_a is taken will correspond to the condition for which the volume in cubic feet per minute is computed. Usually v_a is based on outside air conditions.

Having determined the quantity of ventilation air, the heat loss of the airplane is added to the heat required to bring the ventilation air to inside temperature conditions to find the size of the heating plant required. Before selecting the size, however, the type desired is usually determined.

STEAM SYSTEM

Most present day commercial transports of the DC-3 type carry their own steam heating systems, complete with boiler, radiator, and automatic boiler water feed systems. Although the details may vary among various operators, an understanding of one type is sufficient to understand all the others. Fig. 3 shows one such system.

The source of heat is the engine exhaust, which causes water in the boiler to be flashed into steam and carried in pipes to the radiator located in the ventilating air stream in the fuselage. Here the steam is condensed back into water and returned to the boiler. As in nearly all present day aircraft heating systems, the ventilating air for heating is introduced by the ramming action of the airplane flying through space. A pressure tap in the steam line head of the radiator provides a source of steam pressure which is indicated on a steam pressure gage in the cockpit and which is led into the pressure regulating valve to control the pressure in the system. Since this is an important and interesting part of the system, it will be discussed in detail.

The pressure regulating valve is a diaphragm or bellows actuated mechanism which responds to excessive pressure by closing, and to insufficient pressure by opening. Mounted on the steam expansion tank, which holds the system's excess water, it controls one of the two openings into the tank and therefore when the valve is closed due to excessive pressure, the steam above the water level in the tank is trapped and, as it condenses, the pressure in the tank decreases and circulating water is forced into the tank thereby decreasing the quantity of steam remaining and decreasing the steam pressure in the system.

FIG. 5. STEAM HEATING SYSTEM EXPANSION TANK

If the pressure becomes too low, the regulating valve opens, admitting steam above the water line in the expansion tank and forcing water from the tank into the system where it flashes into steam and raises the steam pressure in the system. The boiler, expansion tank, and steam radiator are shown in Figs. 4, 5, and 6.

The principal advantage of the steam system is safety. Its disadvantages are: (1) its weight is approximately 108 lb or 25 lb more than an equivalent installation of an internal combustion type of heater, (2) maintenance is excessive (average boiler life is short and excessive time is consumed in changing them), and (3) a quality that can only be called general cantan-

kerousness. The steam heating system was responsible for 248 mechanical flight delays in one year!

EXHAUST HEAT TO AIR HEAT EXCHANGER

The simplest type of heating system is the exhaust heat to air heat exchanger, commonly called the heat exchanger. Here again, the source of heat is the engine's exhaust which gives up some of its heat in the exchanger to the ventilating air. The heated air can be diluted with cold air if desired, and

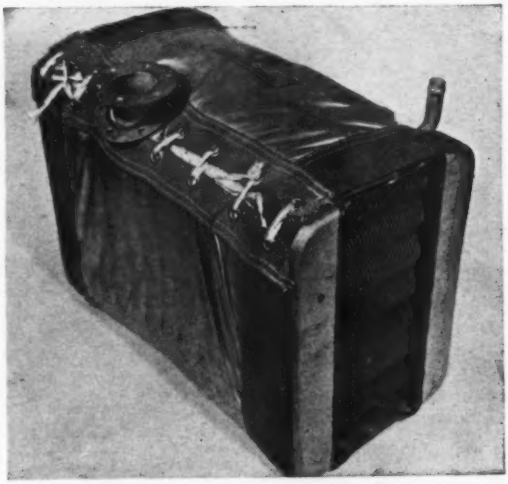


FIG. 6. ALUMINUM AIRCRAFT STEAM RADIATOR

the mixture is delivered to the cabin. If no heat is desired, the heated air is discharged overboard, for at all times outside cooling air must be circulated through the heat exchanger to prevent over-heating of the metal.

This system is simple and practically infallible. Its disadvantages are: (1) installation weight (comparable to the steam system), control difficulties and excessive temperatures, and (2) the rigorous inspections and periodic checks required to prevent the possibility of carbon monoxide escape into the cabin. A further disadvantage, which is also applicable to the steam system, is that these systems cannot be utilized as a source of heat for the cabins while the airplane is on the ground.

The heat exchanger system is used on many military models, as well as on transports operated by several commercial airlines. Its adherents are quick to point out that, as long as the engines are in operation, a source of heat is readily available, that no extra fuel or water are required, and that operation

of the system requires no special technique. Certainly no one who has given any thought to the matter can help but be impressed by its obvious simplicity.

It must be pointed out, however, that although great strides have been made in the metallurgy of heat exchangers, the manufacturing problems are

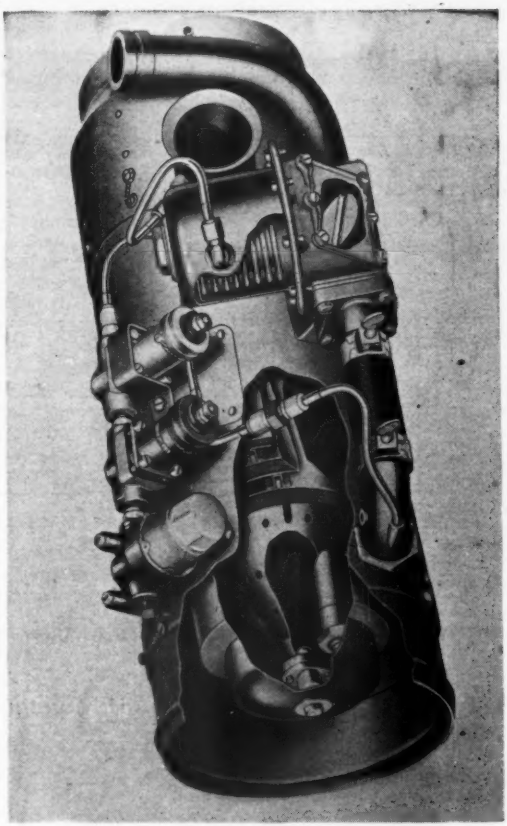


FIG. 7. GASOLINE TYPE COMBUSTION HEATER

most difficult. For example, during takeoff great power is developed by the engines resulting in extremely high exhaust temperatures, at a time when only very small air flows over the heat exchanger exist. During cruise and descent, on the other hand, low power (and therefore low exhaust temperatures) and high air flow rates exist. Under both conditions, satisfactory

(but not excessive) air discharge temperatures must be maintained, and the exchanger must be kept cool to prevent overheating. The lack of cooling air imposes a terrific strain on the heat exchanger, resulting in the manufacturing difficulty previously mentioned.

INTERNAL COMBUSTION HEATER

The newest type of heating system used in airplanes, and the lightest in weight, is the gasoline or combustion heater shown in Fig. 7. There are

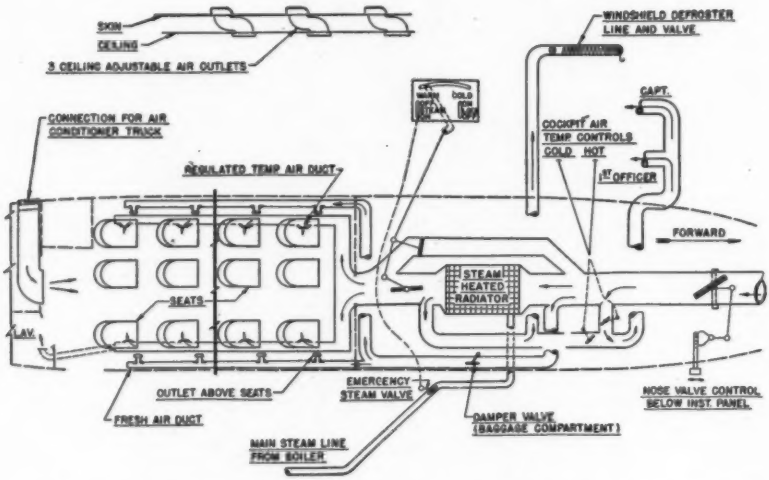


FIG. 8. VENTILATION AND HEAT DISTRIBUTION AMERICAN AIRLINES DC-3 AIRPLANE

several designs on the market, but the ones most widely used today have the following features in common:

1. The fuel used is gasoline, which is pumped up to an atomizing type of spray nozzle under pressure.

2. The nozzle is similar to that used on gun type oil burners and delivers a fine spray of fuel into the combustion chamber where, mixed with the correct amount of air for combustion, it is ignited, burned, circulated through the heater, and discharged overboard.

3. The type of ignition may be either a hot wire or a spark plug.

4. While passing through the heater, the hot products of combustion surrender their heat to the ventilating air which carries it it to the desired part of the airplane.

One 200,000 Btu per hour gasoline-fired heater has an exhaust temperature between 800 and 1,000 F, and has a high efficiency. The unit obtains high

efficiency by means of an enormously high heat transfer rate per square foot of heating surface. The high velocity air passing over the heating surface causes sufficient turbulence and speed of air to reduce the surface coefficient of heat transfer and increase the conductance. The life of these units is relatively low, as compared to a home heating boiler, but is quite satisfactory for aircraft usage where periodic replacements and careful maintenance are readily achieved.

The advantages of the combustion type heater are: (1) its light weight (a 100,000 Btu heater can be easily carried in one hand), and (2) its ready heat availability—heat can be generated on the ground without the costly and troublesome process of operating the engines. Its disadvantages are: (1) its newness—all the troublesome quirks and bugs have not been removed yet, (2) the high maintenance and inspection required to guard against carbon monoxide fumes being introduced into the ventilating air (although the quantity of carbon monoxide is smaller than that found in the case of the heat exchanger), and (3) the fact that raw fuel must be piped to the heaters.

In making a selection of the proper type of heater, the following facts must be borne in mind. Each installation is different, and whereas in one type of airplane the exhaust heater is thoroughly satisfactory, it may be completely wrong for another type of plane. While the steam system seems to be obsolescent, it is entirely possible that new improvements or new modifications on this idea may result in the extensive use of this type of system.

Between the combustion type and the heat exchanger type, it would seem that the advantages of the combustion type would outweigh those of the exchanger—but reliability and infallibility is a big inducement and, unless the manufacturers of combustion heaters can eliminate the numerous sources of failure of their units, the combustion heater may prove to be an unwise choice. In this connection, it would be well to point out that the manufacturers of this latter type of heater have been making enormous strides toward improving performance of their product.

DUCT SYSTEMS

Three types of distribution systems are commonly in use. A sketch of the type used on DC-3 Airliners appears in Fig. 8. Air is brought into the airplane by the impact pressure created by the airplane flying through the air. The ventilating air is forced over the heater, into a mixing chamber where it is diluted with cold air as required and then carried down the side walls of the airplane near the floor and discharged through outlets toward the aisle. Although this system has been in use for some time, it is not completely satisfactory and occasionally passengers notice drafts around the ankles. The principal reason for dissatisfaction probably lies in the fact that the air is admitted into the cabin too near the occupants.

In order to overcome this difficulty, the second or overhead distribution system was evolved. In this system the air, instead of being forced down the side walls of the airplane near the floor, is directed aft along the center of the ceiling and is discharged downward through diffuser type outlets appropriately placed. This system also has not been found ideal because the hot air admitted so near the ceiling becomes too cold in extreme weather before it reaches the

floor unless it is forced downward in a jet in which case it becomes quite uncomfortable for those passengers sitting near the outlet.

Very recently a new system was introduced—so recently, in fact, that complete information as to its performance is not available at this time. This system embodies the use of heated walls. The hot air is passed along the side walls, with branches being led into the air space between the skin and the inner lining of the airplane. In this space, heated air is carried up the side and discharged through a grille near the top of the airplane. Theoretically this system should prove superior to all other distribution systems yet introduced, and it is being watched eagerly by all operators.

Whatever type of distribution system is in use, a source of cold air should be provided for the passengers for combating airsickness and providing ventilation during hot weather flying. The cold air is usually introduced by means of individually adjustable outlets, so that each passenger can control the cold air as desired. Precautions must be taken to prevent any passenger from unwittingly annoying a neighbor with drafts. This is usually accomplished by limiting the travel on each outlet, or by using outlets which will dilute the air before it has travelled sufficiently far to be objectionable.

AUTOMATIC CONTROL

In addition to the various safety controls necessary with each type of heating plant, there is a fertile field in the airplane for automatic temperature control devices. The speed at which the airplane travels, the broadness of the territory over which it flies, and the huge air change rate necessitated by the small size of the cabin all contribute to a rapidly changing cabin temperature within an almost incredibly short time. Although present practice in the DC-3 airliner is limited almost entirely to manual control, several installations of automatic temperature control systems have been made and operators agree that such a system is almost a necessity if cabin temperature is to be maintained constant.

An interesting control system is the electronic one, in which the three controllers, the thermostat, the hot air compensator, and the cold air compensator, are made up of many turns of wire varying in resistance with the temperature. These units are wired into a circuit similar to the Wheatstone bridge, and any variation in the temperature of one of the elements causes a change in resistance inducing a current of very small magnitude. This current is amplified in the electronic amplifier controlling a special motor which opens or closes a damper. The use of the three elements produces an almost constant temperature and instant anticipation of changing heat requirements.

Excellent results with even greater reliability and less weight have also been obtained by the use of a simple electric thermostat connected to an electric proportioning damper motor. Here the change in cabin temperature is more noticeable since no anticipating element is present, but the added simplicity of the system is quite advantageous. All such equipment must be carefully mounted, since there is considerable vibration which would otherwise be quite troublesome to the thermostatic control. Thus a hovering thermostat would be jarred into an open or a closed position it might not otherwise take.

Before the subject of automatic control is dropped, it would be well to point out that the two separate *air zones* on present day aircraft must be treated separately. It is inadvisable to try to control the cockpit temperature from the cabin thermostat, or that in the cabin from one in the cockpit. Rather, in most transport airplanes today, the cockpit has its own system of manual control which the pilots can operate as required. An important feature of cockpit ventilation is the provision for supplying hot air for use in windshield defrosting.

FUTURE CONSIDERATIONS

As aircraft operation improves and as comfort standards improve aboard the aircraft, many new problems will be raised. One of the most important of these is the supercharged or pressurized cabin. This will logically fall in the hands of the aeronautical air conditioning engineer, since he is responsible for the ventilation and thermal comfort of the passengers and crews.

The purpose of pressurizing an airplane is to increase the density of the air in the airplane to achieve an effective altitude in the cabin considerably lower than the altitude at which the airplane is flying. This will enable the airplane to fly at an altitude of 20,000 ft for example, with the crew and passengers suffering none of the attendant disadvantages usually associated with high altitude flying. A recent comment made by a crew member of a pressurized B-29 revealed that he felt less fatigued after a 14 hour flight on a B-29 than he normally felt on a 7 hour flight in other airplanes.

Many problems are still unsolved. For example, the problem of cabin pressure control, cabin temperature control, ventilation control, and atttendant problems will have to be worked out in the future, although some of the details are known now. Many problems not yet dreamed of will arise in connection with pressurizing systems. But all these concern the aeronautical heating and ventilating engineer. To solve them, he will be required to coordinate information from all sources—and much of his knowledge, not only on pressurization but on all aspects of comfort in the air, must be based on information secured from the rest of the heating and ventilating industry.

DISCUSSION

MEYER FAKTOR,[2] Chicago, Ill.: In accordance with what has been shown, it seems to me that the whole thing is a gravity system. It does not look as if it is operated by a blower.

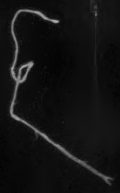
MR. BROD: Perhaps I should have made that a little bit clearer. The airplane is flying through the air at high speed and the impact pressure of the airplane at about 129 mph is about 7 in. of water pressure. Actually, we get sufficient pressure in our duct system just by the ramming action of the airplane flying through the air to operate at a pressure in the duct system of about 5 in. The pressure loss through the ventilating air passages of combustion type heaters at some temperatures is in the neighborhood of 3 to 5 in. of static pressure.

J. H. CARBONE, New York, N. Y.: Mr. Brod, you mentioned something about the heating being objectionable in so far as the temperature at the floor outlets is con-

[2] Reliable Sheet Metal Co.

cerned. I thought I would ask you what temperatures were used in the outlets at the seats.

MR. BROD: Unfortunately the temperature varies since, in the steam system, the heat loss in the duct is rather great. At the outlet nearest the heater we have outlet temperatures approximately 130 to 140 F in severe weather, while during milder weather, we do not need to maintain such a high discharge temperature, and it may be as low as 90 F. As we move aft in the cabin, due to the heat loss from the duct, the temperature drops until at the last outlet it probably is 30 or 40 deg lower than at the first outlet.



1289

No. 1289

EFFECT OF BRANCH TAKE-OFF DESIGN ON NOISE IN VENTILATING DUCT SYSTEMS

By Oscar Imalis,* F. C. Houghten,** Pittsburgh, Pa., and Clark M. Humphreys,† Cleveland, Ohio

JUST PRIOR to and during the war the A.S.H.V.E. Research Laboratory carried on a number of research projects for the U. S. Navy, Bureau of Ships. The results of this work have recently been released for publication. This paper is an abstract of a longer report submitted to the Navy under the same title.

Test Set-up

The arrangement of the system used for this study is shown in Fig. 1. Drawings of the several sections of the system indicate the location of straighteners, orifice, and pitot tube stations. Air flow was controlled by terminal dampers at the ends of both the trunk and branch ducts, and a damper at the fan inlet. The center line of the duct was located at a height of 10 ft 0 in. above the floor and 3 ft 8 in. from the ceiling. An axial-flow type fan was used to move air through the system.

Sketches II and III of Fig. 1 show details of the two types of branch fittings studied. It should be noted that the sizes of branch and continuing trunk ducts are different for the two fittings.

Measurement and Analysis of Sound

Sound measurements were made with an Electrical Research Products, Inc., No. RA 245 sound level meter and four non-directional microphones. During the latter part of the study, a No. RA 243 analysis filter set was used in conjunction with the sound meter. With this filter, it was possible to study the noise level in any one of 14 frequency bands.

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† Senior Engineer, A.S.H.V.E. Research Laboratory.

Presented at the 52nd Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January 1946.

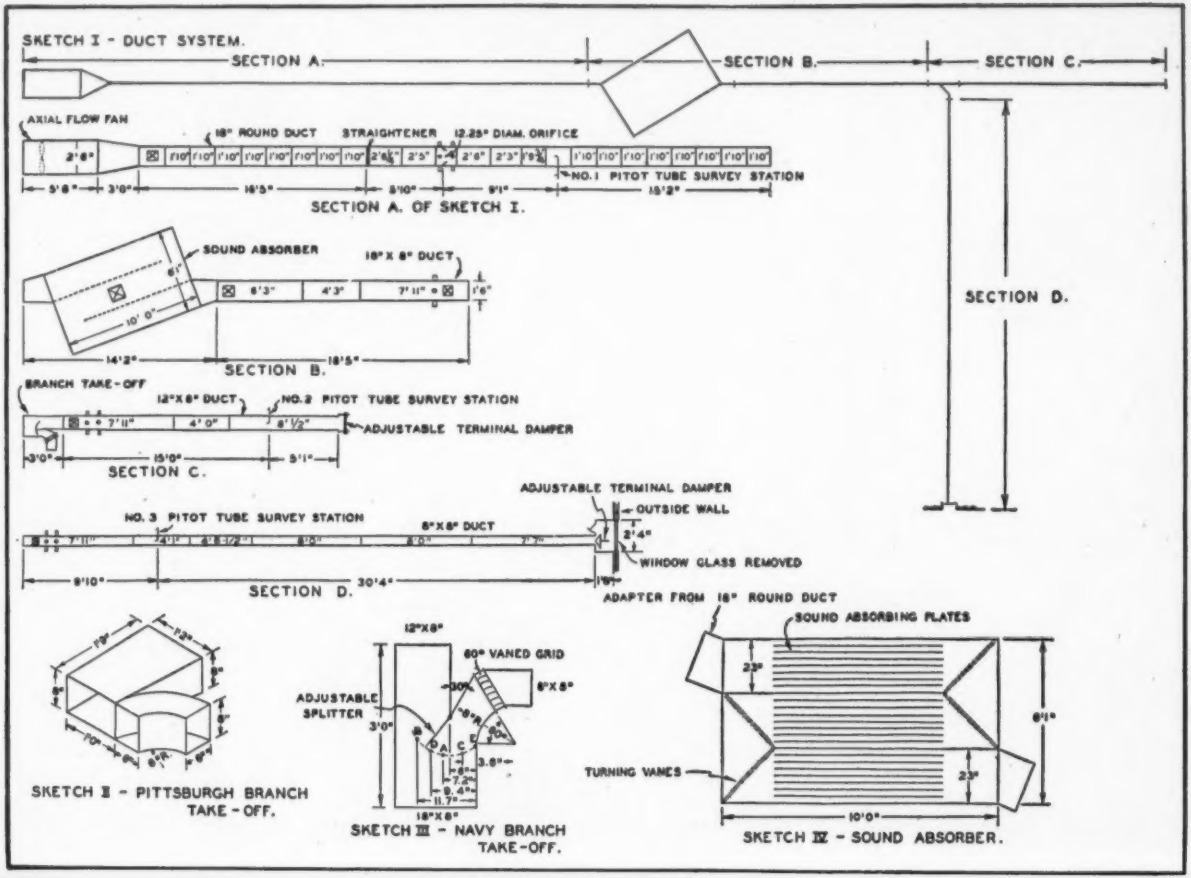


FIG. 1. LAYOUT OF DUCT SYSTEM

The microphones were connected to the sound meter and filter by shielded conductors of sufficient length that the meter and filter could be kept at a central observation station and the microphones moved to any desired point along the duct system. By using four microphones properly located in advance, it was possible to make comparative observations with little or no disturbance, and in a very short interval of time. Most of the actual testing was done at night after other noise making operations in the building were shut down, so that observations could be made with low and uniform ambient conditions.

Fig. 2 is a plan view of the room in which the study was made and shows the duct set-up and 18 numbered microphone stations. All of the sound levels herein reported were taken at stations 1, 2, 3, and 6. At stations 1, 2, and 3 the microphones were located on top of the duct and picked up the noise

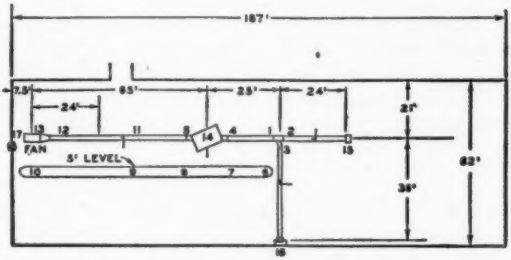


FIG. 2. PLAN VIEW OF TEST SET-UP IN THE LABORATORY SHOWING LOCATION OF OBSERVATION STATIONS

levels in the approach duct, continuing trunk duct, and branch duct, respectively. The microphone at station 6, located 5 ft above the floor and 5 ft below the duct, was used throughout the study to determine ambient noise levels.

In the early tests it was difficult or impossible to separate the fan noise transmitted along the air stream from the noise created at the branch take-off. To reduce the fan noise level, a sound absorber, constructed as shown in Fig. 1, was installed between the fan and the branch take-off.

NOISE LEVEL IN BRANCH TAKE-OFFS

The first three series of tests were made with the Pittsburgh branch take-off. (See Sketch I, Fig. 1.) The next series, made with the Navy branch take-off with the splitter in the neutral position *A*, indicated somewhat higher noise levels than had been found with the Pittsburgh take-off. The order of difference is indicated in Table 1.

In a fifth series of tests with the Navy take-off, the splitter damper was first set in the neutral position *A*, and the terminal dampers were adjusted to give the same velocities in the trunk and branch ducts. With the terminal dampers remaining in this position, the splitter damper setting was then varied and the resulting sound levels at the take-off observed. The results of some

FIG. 1. LAYOUT OF DUCT SYSTEM
TAKE-OFF.

of these tests are given in Table 2. It is evident from these data that movement of the splitter damper in either direction from the neutral position resulted in an increase in noise level in at least one of the three microphone positions.

METHODS OF REDUCING NOISE IN BRANCH TAKE-OFFS

A careful analysis of the data indicated that most of the noise observed with the Navy take-off was created back of the splitter in the continuing trunk

TABLE 1—RELATIVE NOISE LEVELS OF PITTSBURGH AND NAVY TAKE-OFFS[a]

TEST NO.	BRANCH TAKE-OFF	VELOCITY, FEET PER MINUTE			NOISE LEVEL IN DECIBELS			
		POS. 1	POS. 2	POS. 3	POS. 1	POS. 2	POS. 3	POS. 6
106	Pittsburgh	3890	3180	3640	88.5	87.5	91.5	79.0
126	Navy	3632	2990	3560	93.0	106.0	96.0	....

[a] Trunk and branch dampers open 4 in. and 8 in. respectively, and fan damper wide open during both tests.

TABLE 2—VARIATION IN SOUND INTENSITY DUE TO VARIATION IN SPLITTER SETTING[a]

TEST NO.	SPLITTER POSITIONS[b]	VELOCITY, FEET PER MINUTE			NOISE LEVEL IN DECIBELS			
		POS. 1	POS. 2	POS. 3	POS. 1	POS. 2	POS. 3	POS. 6
162	A	4231	3654	3665	118.0	114.0	112.0	85.0
163	B	3635	2530	4120	117.0	128.0	118.0	87.0
164	C	3882	4235	2390	118.0	119.5	112.0	85.5
165	D	3998	3200	3812	118.5	121.0	115.0	85.0

[a] Trunk and branch dampers open 5⅝ in. and 8 in. respectively, and fan damper wide open during all tests.
[b] See Fig. 1, Sketch II.

duct, position 2, and that it was caused by turbulence in the bulk of relatively stationary air at that point. A number of tests were made with various modifications in the design of the splitter to determine methods of reducing this turbulence and the resulting noise. Some of the splitter designs tested and the results obtained are given in Table 3. From these results it is evident that the noise created at a splitter damper may be reduced appreciably by proper streamlining. However, this study was not carried to the point of developing an adjustable streamlined mechanism which would be capable of functioning as a splitter.

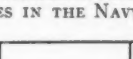
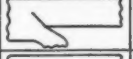
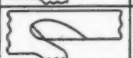
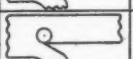
All of the tests recorded in Table 3 were made with the same terminal damper settings and at the same fan speed. The air delivery was, therefore, dependent only upon the frictional resistance of the system. It is interesting to note that, in general, as the noise level was reduced, the air delivery in-

creased. In test No. 251, which indicated the lowest noise level, the air delivery was approximately 16 per cent greater than in test No. 236.

EFFECT OF MICROPHONE ARRANGEMENT ON NOISE LEVEL

It was evident throughout the testing program that the actual numerical values of sound level obtained were dependent to a large extent on the location and general arrangement of the microphones. The sound levels observed during any particular series of tests can only be considered comparative, and

TABLE 3. NOISE LEVELS AND ACCOMPANYING VELOCITIES FOR VARIOUS FLOW REGULATING DEVICES IN THE NAVY BRANCH TAKE-OFF

Test No.	Splitter, Branch Take-off Arrangement		Velocity in Feet Per Minute			Noise Level in Decibels -- Total			
			Pos. 1	Pos. 2	Pos. 3	Pos. 1	Pos. 2	Pos. 3	Pos. 4
236	Splitter held in Position "B"		3453	2475	4030	101.1	126.1	105.7	66.7
238	Unbowed sheet at 10° to wall		3687	2565	3790	104.3	125.2	107.9	65.1
239	Convex-bowed sheet 3⅛ 3/4-in. attached to splitter "B" position		3817	2965	3470	98.9	113.8	105.6	63.2
240	Streamlined casing over "B" position of splitter		3762	3040	3580	97.2	108.3	102.2	68.2
241	8.7-in. dia. cylinder		3646	2905	3680	101.6	120.8	107.4	67.6
250	8.7-in. cylinder and 65-in. rigid tight fitting sheet, sealed with plastic cement		3788	3050	3660	97.8	117.5	103.0	66.2
251	Same as Test No. 250, except last 3 in. of rigid piece curved against cylinder		4006	3170	3560	97.5	99.3	103.2	65.8
252	Same as Test No. 251, except 1/4-in. ridge of Permoplast		3729	2975	3690	97.6	121.0	103.3	66.0

since the microphone arrangement was changed several times during the study, not all series of tests are directly comparable. This is evident in a comparison of the results of test No. 126 given in Table 1 and test No. 162 given in Table 2. Much of the difference in sound levels observed in these two tests is due to a change in microphone arrangement.

A series of tests was made to determine more accurately the effect of microphone location and arrangement on total sound level and also on the sound levels in the several frequency bands. Readings were taken with different types of openings in the duct and with different housings over the microphones.

Fig. 3 is presented as an example of the method of analysis used in these tests. This figure shows the results obtained with an open 4 in. x 4 in. hole

in the duct, no hole, the duct perforated with nine ¼ in. holes, and with 32 ¼ in. holes. In these four tests, the microphone rested on a ½ in. thick piece of sponge rubber laid on top of the duct, and was covered with a 5 in. x 12 in. x 8 in. deep box lined with ½ in. sound absorbing material.

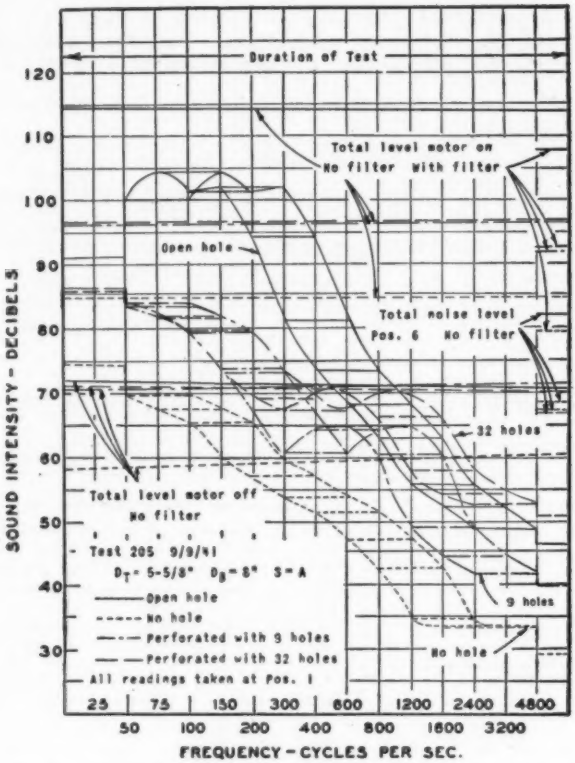


FIG. 3. NOISE LEVEL COMPARISON FOR VARIOUS OPENINGS IN DUCT

In the graphic representation of frequency analysis, the total noise intensity within a band is shown by a horizontal line and the beginning and ends of these lines are connected by continuous curves. There are no overlapping bands in the frequency range below 50 and above 4800 cycles. The total noise levels, titled *Motor on, no filter, Motor off, no filter, Position 6,* etc., were taken at the beginning and end of each test, and are a check of constancy of noise conditions during the test; they do not refer to frequency

variation but to the *duration of test* time scale. D_t, D_b and S refer to terminal trunk damper, terminal branch damper, and splitter setting, respectively.

The sound intensity distribution with respect to frequency as shown in Fig. 3 is typical of the results obtained throughout the study. In every test, the maximum intensity was found to be in or below the 75 to 150 cycles per second frequency band.

SUMMARY

1. The noise created by the splitter in a branch take-off arises from the turbulence of a relative stationary mass of air back of the splitter. The noise can be reduced by properly streamlining the splitter.

2. Noise level values obtained in a study of this kind vary widely depending upon the arrangement and location of the microphone, and the type of opening between the air stream and the microphone. Universally accepted standards of sound measurement techniques are badly needed in many phases of the ventilating and air conditioning industry. Until such standards are available, the proper interpretation of sound measurements, and the correlation of the results of various investigators in the field, will be difficult or impossible.

3. The greater part of the noise created in a duct system is of relatively low frequency. Maximum intensities in this study occurred at or below 150 cycles per second.

4. This study was confined to the performance of branch take-offs only, and does not provide a means of determining how much the noise level in a room may be increased due to a certain noise production in the duct system.

DISCUSSION

MEYER FAKTOR,[1] Chicago, Ill.: What will happen if you take all those splittings and insulate them with celotex, or some other insulating material? From my experience in the past I solved a problem—I cured the noise.

DR. IMALIS: We tried to line the ducts, but did not have much success. The noise is created by small turbulent motion of the air that propagates. This is also the reason why we have the biggest noise at a frequency of around 100 to 150. We first tried to use a large box as a sound absorber. This was unsuccessful because, as you have seen in the first diagram, the sound absorber was located between the fan and the damper. However, the sound, which we had thought was created in the fan, was merely created by the damper and by turbulences due to the sharp edges.

In one of the slides you have seen a cylinder, to which we fastened a plastic ridge, about 1 in. high. This ridge created an additional sound of around 15 or 20 decibels. This is a large figure. However, as we took off this plastic ridge the sound level dropped immediately. This was an experimental test to prove that the main source of sound is the turbulence created in the air. With the streamlined fitting and the smooth surface we can get better results. However, our sound absorber box, even with a lining, did not get any results.

C. M. HUMPHREYS, Cleveland, Ohio: May I first thank Dr. Imalis for his remarks, and then introduce him to you as the one who was responsible for these tests at the Laboratory.

B. M. BROD, La Guardia Field, N. Y.: I would just like to raise the question as to whether the reason that you were not successful in eliminating the noise by

[1] Reliable Sheet Metal Co.

means of sound absorbers of porous materials was due to the low frequency of the noise. Are not the best results in reducing noise with sound absorbers obtained at high noise frequencies and for low frequencies is not the best noise reduction accomplished by means of very heavy mass? Such an absorber, of course, would not be practical in a duct system.

CYRIL TASKER, Cleveland, Ohio: I think there is one point which perhaps has not been brought out, and which was deliberately omitted by Mr. Humphreys, because of lack of time. That is, that quite obviously from these, and other studies, the location of the microphone and its orientation with regard to the equipment under test is by no means standardized, and therefore, it is extremely difficult to make comparisons between tests that you may see reported because of that very factor.

It seems to us that some universally acceptable standards for noise testing are very much to be desired if we are going to be able to correlate the results of the investigations made by various workers in the field. We found that to be so when some of us went to the Brooklyn Navy Yard and saw some of the means being used for testing the noise level of fans and other pieces of ventilating equipment. I know that some of you are only too well aware of the difficulties entailed by using different types of sound-measuring equipment. One of our jobs is to try to overcome these difficulties, and recently we have had some earmarked funds made available to us, which we may use to conduct some studies. Some while ago the Committee on Sound Control gave me the job of trying to arrange with the other authorities in the field, some standardization of sound-measuring equipment. This is something of a tall order, but I think if we are going to be able to get comparable results, and results which anyone in the field can understand, we are going to have to try to get more standardization, both of equipment and testing procedures, than we have at the present time.

1290

No. 1290

PHYSIOLOGICAL RESPONSE OF SUBJECTS EXPOSED TO HIGH EFFECTIVE TEMPERATURES AND ELEVATED MEAN RADIANT TEMPERATURES

By CLARK M. HUMPHREYS,* CLEVELAND, OHIO; OSCAR IMALIS,** AND CARL GUTBERLET,† PITTSBURGH, PA.

JUST PRIOR to and during the war, the A.S.H.V.E. Research Laboratory, then located in Pittsburgh, Pa., carried on a number of research projects for the U. S. Navy, Bureau of Ships. The results of this work have recently been released for publication. This paper is an abstract of a longer report submitted to the Navy under the same title.

OBJECTIVE AND SCOPE

The objective of this research project was to determine the physiological reactions of men exposed for four hours to hot atmospheres and surrounded by hot surfaces. The study included tests at Effective Temperatures ranging from 80 to 95 deg, with relative humidities of 30, 50, and 70 per cent, and mean radiant temperatures up to 40 deg above the dry-bulb temperature of the air.

DEFINITION OF TERMS

The *Effective Temperature* (ET) is an arbitrary index which combines into a single value the effect of temperature, humidity, and the movement of air, on the degree of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation of warmth. The full range of conditions included in this study is shown on the Effective Temperature Chart, Fig. 9.

The *Mean Radiant Temperature* (MRT) is another factor, in addition to those included in the definition of Effective Temperature, which affects man's feeling of warmth. It is defined as the uniform surface temperature of an

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enclosure at which one would exchange the same amount of energy with the enclosure by radiation as at the non-uniform wall temperatures found in practice.

The MRT Elevation, as used in this report, is the excess of the mean radiant temperature over the dry-bulb temperature of the air.

DESCRIPTION OF THE TEST ROOM

Tests were conducted in one of the Laboratory's psychrometric rooms, to which certain alterations were made to adapt it to this study. To obtain elevated mean radiant temperatures, radiant panels were installed on all four walls of the room. Each panel consisted of a metal sheet, to the back of which a copper coil was soldered. The panel temperature was controlled by varying the pressure of the steam supplied to the coil.

To reach the higher MRT elevations desired for the tests, it was necessary to install reflective surfaces on the floor, on the ceiling, and on the walls below the radiant panels. High MRT elevations were more difficult to secure at the higher relative humidities because of the semi-opaqueness of water vapor to radiant heat.

Air from conditioning equipment in an adjoining room was delivered to a plenum chamber which extended over the greater part of the ceiling of the test room. From this chamber it was discharged above the reflective ceiling and entered the occupied area through an opening extending along all four sides of the room. A part of the air was recirculated, and the remainder was exhausted to the surrounding rooms.

INSTRUMENTS

Globe Thermometer

The mean radiant temperature of the room was determined by the use of 6 in. diameter unheated globe thermometers located 48 in. above the floor.

The MRT readings were calculated from the globe readings, air temperature, and air velocity, using Bedford's equation.[1]

Hot-wire Anemometer

The air movement in the test room was measured with a hot-wire anemometer.

Aspirated Thermocouple

The dry bulb temperature of the air in the test room was measured by a thermocouple located in a thermos bottle through which room air was drawn by a small fan. The temperature thus obtained was used in the calculation of the room MRT as mentioned above. Both the thermos bottle and fan were completely covered with aluminum foil to reduce the effect of radiation.

Thermocouples

All thermocouples used in these tests were made of calibrated No. 36 copper and constantan wire. A precision-type potentiometer was used to measure the emf of the various thermocouple circuits.

[1] T. Bedford and C. G. Warner, *Journal of Hygiene* (London), Vol. 34, 1934, pp. 458-473.

Clinical Thermometers

Clinical thermometers were used to determine rectal and oral temperatures of the subjects.

Wet Bulb Thermometers

Wet bulb temperatures were taken with a mercurial thermometer, the bulb of which was covered with a water saturated wick. The thermometer was inserted into the exhaust air duct.

SUBJECTS

The study logically divides itself into two parts because of the fact that two groups of subjects were used. The first and major part of the study, which included approximately 75 per cent of the tests, was made with a group of 36 male civilian subjects between 17 and 22 years of age. The latter 25 per cent of the tests were made with a group of seven enlisted men from the Navy who were assigned to the Laboratory to serve as subjects. Except for a later discussion of acclimatization, this paper will deal only with the first group of tests.

As mentioned previously, a total of 36 civilian subjects were used. However, the great bulk of the data was obtained with eight subjects who were selected from the larger group during the early tests. All of these eight subjects were 18 years of age. Before being accepted, each subject was required to pass a physical examination based on standards for Navy enlistment. The basal metabolism of each subject was also determined as further evidence of good health.

TEST PROCEDURE

During the tests the subjects wore regulation Navy clothing issued to men working in hot spaces, consisting of sleeveless shirt, trousers, socks, and shoes.

Before entering the test room, subjects were required to remain at rest for at least one hour in a control room maintained at 77 deg ET. During this time, a light lunch was served for the subjects, all of whom consumed approximately equal amounts. The purpose of this control room was to eliminate the effect of varying exertion or weather en route to the laboratory, and thus to make sure that all the subjects were at equilibrium with a common environment. At the end of the equalizing period, the subjects passed from the control room to the nearby test room, which had previously been brought to the desired conditions.

To simulate conditions in the fireroom of a ship, it was considered desirable to have the subjects engaged in light work of a type which would keep them alert and attentive throughout the testing period. To accomplish this, a definite program of activity was planned and carried out with the aid of a simple chance machine. As the activity of the subjects in tests of this nature has an important influence on the results obtained, the operation is described in detail.

Each subject was given a marble and six small numbered weights of the same distinctive color. Beside the chance machine was a small table on which

six numbered areas were defined. Each subject dropped his marble into the chance machine, which was so constructed that the marble would be returned in from one-half to five minutes. When a marble emerged, its owner immediately threw two dice, one of which indicated which one of several weights should be moved, the other, where it should be placed on the table. When the task was completed, the subject recorded the time and move made and again dropped the marble into the machine.

There was only one chair in the test room for the five subjects. The subject completing a move at the machine took the chair, and the others progressed in a pre-arranged order to new locations marked on the floor of the room.

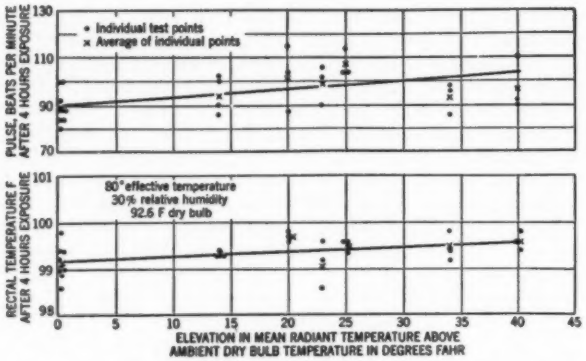

FIG. 1. EFFECT OF MEAN RADIANT TEMPERATURE ON PULSE AND RECTAL TEMPERATURE FOR 80 DEG ET AND 30 PER CENT RH

Thus each man was seated, on the average, 20 per cent of the time. This rotation also assured exposures of equal severity for all subjects.

Tests were normally of 4-hour duration to conform to the customary Navy watch period. However, if a subject's pulse exceeded 140 beats per minute, or his rectal temperature 102 F, he was removed from the test.

Subjects were permitted to drink as much water as they desired, but were required to drink it at least 10 min before temperatures were to be taken. In accordance with instructions from the Navy, a sodium chloride concentration of 0.15 per cent by weight was maintained in the drinking water. The weight of water consumed by each subject was recorded.

COLLECTION OF DATA

The following physiological data and observations were taken during each test: (1) Body temperature, rectal and oral; (2) Pulse rate; (3) Weight of subjects before and after test; (4) Weight of water consumed and urine expelled by each subject; (5) Blood pressure.

Blood pressure was taken before and after each 4-hour test. Body temperatures and pulse rate were taken at 15-min intervals.

The following physical data were taken throughout each test: (1) Dry bulb temperature; (2) Wet bulb temperature; (3) Globe thermometer readings; (4) Air velocity.

PRESENTATION OF DATA

Tests were made at 30, 50, and 70 per cent relative humidity. The data obtained at 30 per cent, showing the effect of environmental conditions on pulse and rectal temperature, are presented in Figs. 1 to 5. In these figures,

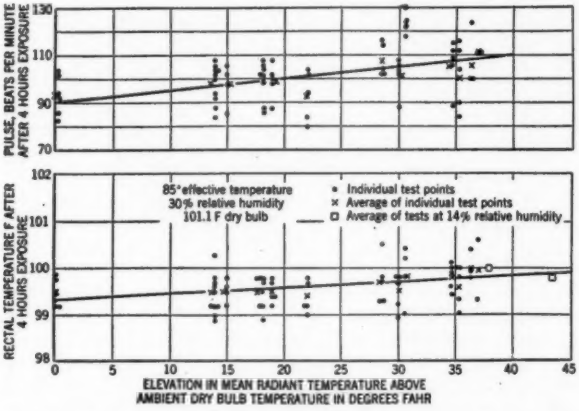

FIG. 2. EFFECT OF MEAN RADIANT TEMPERATURE ON PULSE AND RECTAL TEMPERATURE FOR 85 DEG ET AND 30 PER CENT RH

each dot indicates the reaction of a single subject in one test. A series of not more than five dots in a vertical line indicates the reactions of the group of subjects during one test. There were never more than five subjects used at any one time; where more than five points appear in a vertical line, it indicates two or more tests made at the same conditions. The average value for each test or group of tests is indicated by an X.

It will be noted that the temperatures indicated by the individual points seldom vary more than 0.75 F from the average temperature as shown by the curve. This is a usual range for such physiological data and is due not only to the physiological inequalities of individuals, but also to variations in the same individual at different times.

The effect of mean radiant temperature on pulse and rectal temperature is shown in Fig. 1 for 80 deg ET and 30 per cent relative humidity. Similarly, Figs. 2, 3, and 4 show these same relationships for 85, 90, and 92.5 deg ET, respectively, with 30 per cent relative humidity.

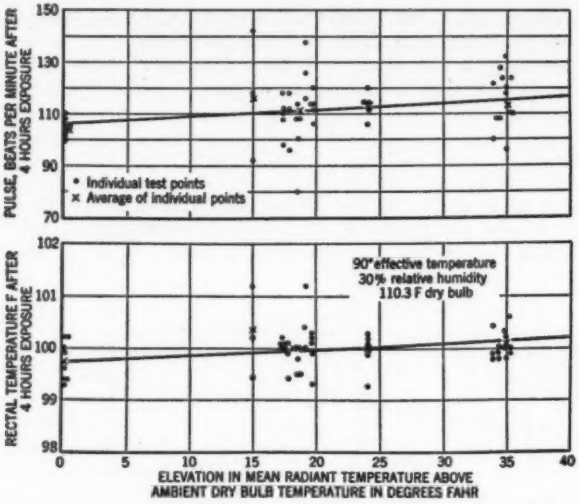

FIG. 3. EFFECT OF MEAN RADIANT TEMPERATURE ON PULSE AND RECTAL TEMPERATURE FOR 90 DEG ET AND 30 PER CENT RH

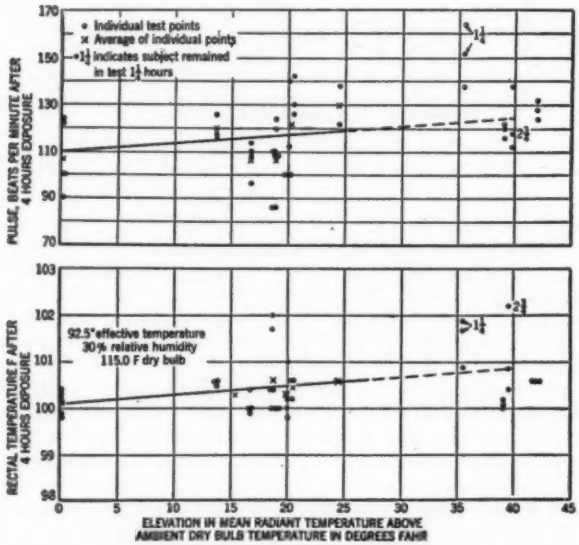

FIG. 4. EFFECT OF MEAN RADIANT TEMPERATURE ON PULSE AND RECTAL TEMPERATURE FOR 92.5 DEG ET AND 30 PER CENT RH

Fig. 5 shows the variation in rectal temperature and pulse with Effective Temperature for 0, 20, and 40 deg MRT elevations. It is constructed, not directly from test data, but from curve points taken from Figs. 1, 2, 3, and 4, all of which are for 30 per cent relative humidity. For convenience, the abscissae are given in corresponding dry bulb as well as Effective Temperatures.

Similar sets of curves were plotted for tests at 50 and 70 per cent relative humidities, but for lack of space have been omitted from this paper. Fig. 6,

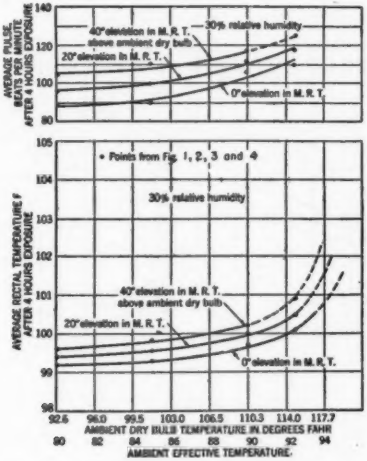
AVERAGE PULSE, BEATS PER MINUTE AFTER 4 HOURS EXPOSURE
140
120
100
80
40° elevation in M. R. T. above ambient dry bulb
20° elevation in M. R. T.
30% relative humidity
0° elevation in M. R. T.
AVERAGE RECTAL TEMPERATURE F AFTER 4 HOURS EXPOSURE
105
104
103
102
101
100
99
98
• Points from Fig. 1, 2, 3 and 4
30% relative humidity
40° elevation in M. R. T. above ambient dry bulb
20° elevation in M. R. T.
0° elevation in M. R. T.
92.6 96.0 99.5 103.0 106.5 110.3 114.0 117.7
AMBIENT DRY BULB TEMPERATURE IN DEGREES FAHR
80 82 84 86 88 90 92 94
AMBIENT EFFECTIVE TEMPERATURE.

FIG. 5. VARIATION OF RECTAL TEMPERATURE AND PULSE WITH EFFECTIVE TEMPERATURE FOR 0, 20, AND 40 DEG MRT ELEVATION

however, presents a summary of all of the tests. The dotted lines in this figure were taken from Fig. 5, and the solid lines from a similar figure for the tests at 50 per cent relative humidity. Tests at 70 per cent relative humidity were made at only the higher Effective Temperatures, and their results are indicated on Fig. 6 by appropriate dots, rather than curves.

Unless otherwise noted on the figures, the plotted points on Figs. 1 to 4, inclusive, represent conditions of pulse and rectal temperature at the end of the 4-hour test periods. It sometimes became necessary during tests at the higher Effective Temperatures to remove a subject from the test room before 4 hours had elapsed, due to excessive body temperature or pulse, or because of exhaustion. In such cases, where the subject had remained in the test room during the greater part of the test period, and where his temperature and pulse had risen at a reasonably uniform rate up to the time of his removal, the data have been extrapolated to determine the probable values for a 4-hour exposure. Such an extrapolated point is indicated by an open circle in the figures. If the

subject was removed because of a sudden and very rapid rise in pulse or temperature, no attempt was made to extrapolate the data. However, in some cases the pulse and temperature for such a subject are indicated on the charts, but beside the dots will appear the number of hours which elapsed before the subject was removed from the test room and the readings taken.

Even under the most favorable conditions the accuracy of extrapolated physiological data is questionable. Including records for only those subjects who were able to withstand the high temperatures for 4 hours, and deleting

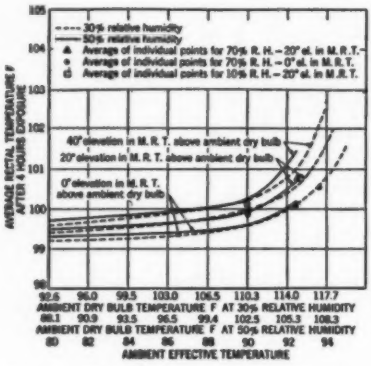
30% relative humidity
50% relative humidity
Average of individual points for 70% R. H. – 20° el. in M.R.T.
Average of individual points for 70% R. H. – 0° el. in M.R.T.
Average of individual points for 10% R. H. – 20° el. in M.R.T.
AVERAGE RECTAL TEMPERATURE F AFTER 4 HOURS EXPOSURE
40° elevation in M.R.T. above ambient dry bulb
20° elevation in M.R.T. above ambient dry bulb
0° elevation in M.R.T. above ambient dry bulb
92.6 96.0 99.5 103.0 106.5 110.3 114.0 117.7
AMBIENT DRY BULB TEMPERATURE F AT 30% RELATIVE HUMIDITY
88.1 90.9 93.5 96.5 99.4 102.5 105.3 108.3
AMBIENT DRY BULB TEMPERATURE F AT 50% RELATIVE HUMIDITY
80 82 84 86 88 90 92 94
AMBIENT EFFECTIVE TEMPERATURE

FIG. 6. VARIATION OF AVERAGE RECTAL TEMPERATURE AFTER 4 HOURS EXPOSURE FOR 30, 50, AND 70 PER CENT RH AND 0, 20, AND 40 DEG MRT ELEVATION

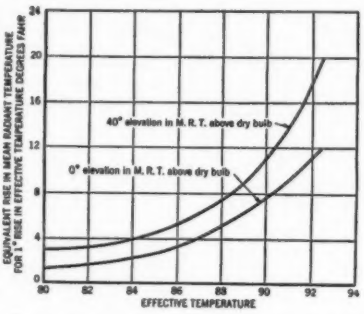
EQUIVALENT RISE IN MEAN RADIANT TEMPERATURE FOR 1° RISE IN EFFECTIVE TEMPERATURE DEGREES FAHR
40° elevation in M. R. T. above dry bulb
0° elevation in M. R. T. above dry bulb
EFFECTIVE TEMPERATURE

FIG. 7. EVALUATION OF EFFECT OF MRT ELEVATION IN TERMS OF EFFECTIVE TEMPERATURE

the rest, also gives results which are open to criticism. In certain respects, the points which would thus be omitted are of particular importance. To try to present the most complete picture possible, curves have been extended on the basis of extrapolated and *short period* points. However, such extensions are shown with dotted, rather than solid lines, and the values indicated by the dotted portions of the curves can be considered only approximate.

Referring again to Fig. 6, it is of interest to note the close agreement of curves and points for the several relative humidities. The small differences shown here become entirely negligible when compared with the spread of points found for each test. This agreement indicates that the results of this project check very closely the earlier work done in the development of the Effective Temperature Charts.

An evaluation of the effect of MRT elevation in terms of Effective Temperature is presented in Fig. 7. It is apparent from this figure that the effect of a rise in MRT on physiological reactions is dependent upon both the Effective Temperature of the surrounding air and the amount of MRT elevation. The curves in the figure were obtained from the slopes of the curves in the preceding figures. For example, at 84 deg ET and 40 deg elevation in MRT, a 4 deg rise in MRT is equivalent to a 1 deg ET change. Similarly,

at a constant ET of 90 deg, and with MRT elevations of 0 and 40 deg, 7.5 deg and 11 deg MRT rises respectively are required to produce a 1 deg change in the Effective Temperature.

In Fig. 8 rectal temperature curves are plotted using Effective Temperature and MRT elevation as coordinates. From this figure it is possible to determine

TABLE 1—SUMMARY OF TESTS MADE WITH CIVILIAN SUBJECTS AND ANALYSIS OF CAUSES OF REMOVAL OF SUBJECTS FROM TESTS

EFFECTIVE TEMPERATURE DEG	90	92.5	93.7	95
Number of tests at this ET	31	21	9	9
Total number of subjects taking part in these tests	143	101	41	42
Number of subjects removed for all causes	17	37	18	33
Percentage of subjects taking part who were removed	11.9	36.6	43.9	78.5

ANALYSIS OF CAUSES FOR REMOVAL OF SUBJECTS FROM TESTS

	Number of subjects and the percentage of the total number of subjects taking part in tests who were removed for			
CAUSE OF REMOVAL	EACH CAUSE (PER CENT)			
A. High pulse	8 (5.6)	25 (24.8)	13 (31.7)	24 (57.1)
B. High rectal temperature	7 (4.9)	6 (5.9)	2 (4.9)	3 (7.1)
C. Exhaustion	2 (1.4)	6 (5.9)	3 (7.3)	6 (14.3)
	The percentage of the total number removed who were removed for			
CAUSE OF REMOVAL	EACH CAUSE (PER CENT)			
A. High pulse	47.0	67.6	72.2	72.7
B. High rectal temperature	41.2	16.2	11.1	9.1
C. Exhaustion	11.8	16.2	16.7	18.2

various combinations of Effective Temperature and MRT elevation which produce the same physiological reactions. For example, a combination of 87 deg ET and 30 deg MRT elevation produce a body temperature of 99.8 F. Following down this body temperature line, it is found that a combination of 91 deg ET and 0 MRT elevation is exactly equivalent to the first set of conditions in its effect on body temperature.

The effect of radiant heat on physiological reactions is much less than had been anticipated. This is particularly true at the higher Effective Temperatures.

ENDURABILITY OF CONDITIONS

Table 1 presents a summary of all tests made on civilian subjects and gives an analysis of the causes for removal of subjects from the test room before the end of the scheduled 4-hour test. It is interesting to note that as the Effective Temperature increased a greater percentage dropped out because of high pulse rate and exhaustion, and a smaller percentage was removed because of high rectal temperature. *Exhaustion* in this analysis includes extreme weakness, sickness, severe headache, etc.

WEIGHT LOSS

An analysis of weight loss by perspiration was made in much the same manner as body temperature and pulse were analyzed in Figs 1 to 6. As

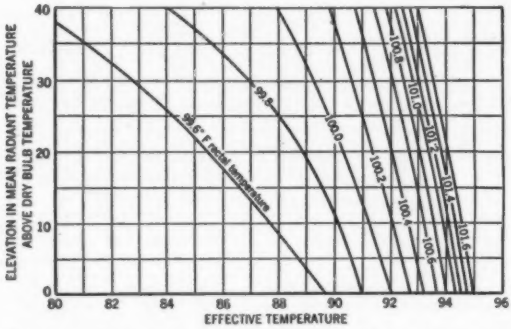

FIG. 8. CHART FOR DETERMINING COMBINATION OF EFFECTIVE TEMPERATURE AND MRT ELEVATION WHICH PRODUCES SAME PHYSIOLOGICAL REACTIONS

might be expected, the weight loss increased with increases in Effective Temperature and MRT elevation. The average weight loss at 92.5 deg ET and 40 deg MRT elevation was found to be about 4.5 per cent of the subject's entering weight. Most of this loss, however, was made up by the water consumed during the test. The difference between the entering and leaving weight of a subject seldom exceeded two pounds.

ACCLIMATIZATION

From the very beginning of the study it was recognized that the ideal subjects for the tests would be men who actually had experience in the fire rooms or engine rooms of naval vessels. It was only because Navy subjects were unavailable that civilian subjects were used for the major part of the work.

One of the chief arguments advanced in favor of Navy subjects was that centering around the subject of acclimatization. It was thought possible that men long accustomed to work in hot spaces would be less affected by the heat.

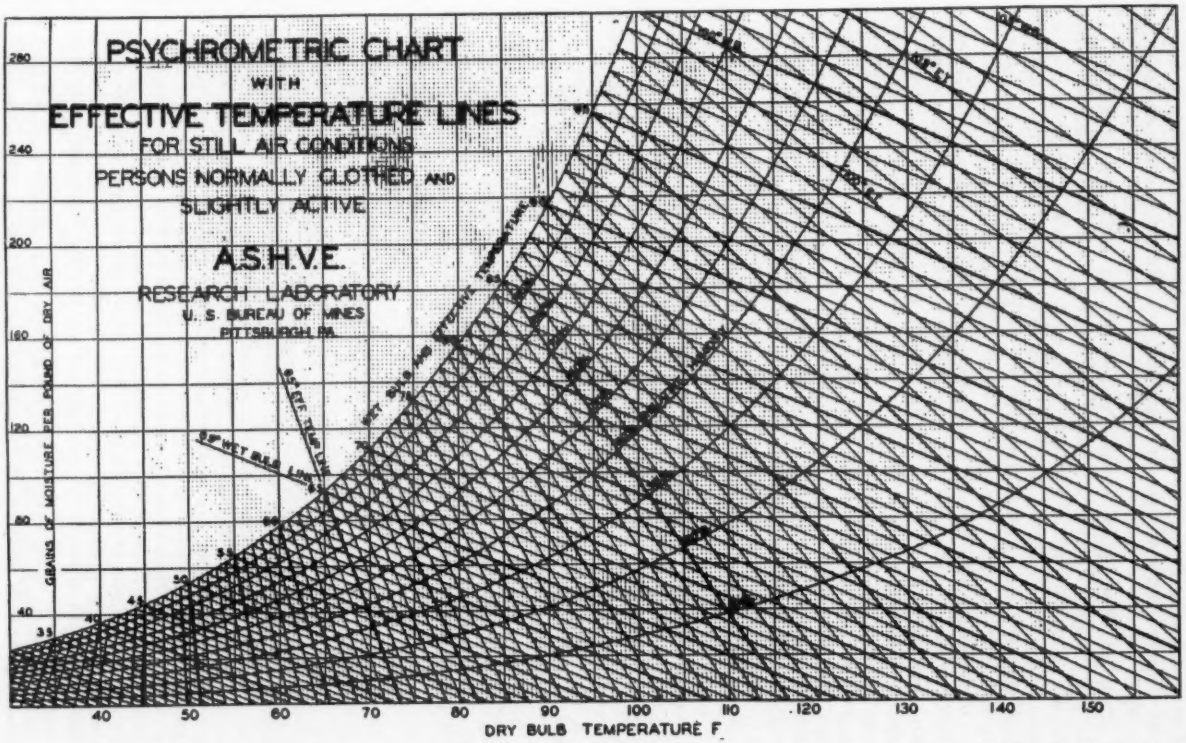

FIG. 9. A.S.H.V.E. EFFECTIVE TEMPERATURE CHART FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE IN STILL AIR

Since the greater part of the study with civilian subjects was made during the cold winter months, the possible effect of acclimatization became doubly important.

Early in May, seven enlisted men from the battleship *Arkansas* were assigned to the Laboratory to serve as subjects. All of the men were from the fire-room and engine-room crews and were sent to the Laboratory the day after their ship docked. However, as it was later learned, their ship had been on patrol duty in the North Atlantic rather than in the tropics, and the conditions under which the men had stood watch were not too severe.

On the assumption that these men were thoroughly acclimatized, they were placed at once in rather severe tests. It was immediately evident that they were not accustomed to such conditions, for their pulse rate and body temperatures reached considerably higher values than those of civilian subjects in tests of equal severity. The group participated in a test each day, and for at least two weeks showed steady and appreciable improvement. At the end of that time they were easily able to remain in the test room for four hours under environmental conditions which none had been able to endure for four hours at the beginning of the period.

SUMMARY

1. Within the range of conditions considered in this study, all variables in a man's environment which affect his physiological reactions may be expressed in terms of Effective Temperature and MRT elevation. When physiological reactions are plotted against Effective Temperature, the results are practically the same for various values of relative humidity.

2. The effect of an increase in MRT elevation on physiological reactions is dependent upon the values of ET and MRT elevation. This is shown in Fig. 7.

3. The effect of radiant heat on physiological reactions decreases as the Effective Temperature increases and becomes surprisingly small as the upper limits of endurability are reached.

4. By repeated exposure to hot environments for periods of 4 hours or less each day, a person can acquire a considerable degree of acclimatization. The acclimatization process may require a period of two weeks or longer.

DISCUSSION

C.-E. A. WINSLOW, New Haven, Conn.: I think some explanations might be made in regard to some of these results. The data are plotted in such a way that they suggest that you would expect to get a progressive increase in the temperature of the thing that is being heated as you increase the heating, which would happen in non-living materials. However, in the case of a living body, as the temperature rises the sweating mechanism begins to operate and cools the body. This temperature regulating effect causes the curves to flatten out at the more extreme conditions.

I would like to suggest that it would be highly desirable to repeat these data on the basis of operative temperature. Of course, it is necessary to know the air movement, or these results are meaningless. With the air movement it should be possible to get much more significant results, I think by plotting the operative temperature, which is the net effect of the air temperature, air movement, and mean radiant temperature.

W. L. FLEISHER, New York, N. Y.: For the last seven or eight years I have been contending that the final temperatures, rather than the rise in temperatures, should

be taken into consideration in the developing of the curve. I have seen some of the charts on which, according to the curves, dangerous conditions existed, necessitating the removal of a subject because of a rise in temperature. In several cases the temperature of the subject removed was no higher than the temperature of the subjects who were left in the test room. This is because the original temperature of the subjects varied more than the particular rise which required their removal. There is a variation in the original temperature of the subjects from 98 to 99.2 F and consequently this represents a 1½ deg greater temperature in the lowest temperature subject when he was removed. The important thing is the final temperature of the subject, and how the temperatures of the various subjects relate to each other, rather than the particular rise of temperature in each case. The ambient room temperature should, under normal conditions, give an even temperature to those who remained in the room and I believe this would be found true if the final temperatures of the subjects had been obtained.

L. T. AVERY, Cleveland, Ohio: Mr. Fleisher raises a question which seems to me so important that we ought to do something about it. In actual observation of the effect on the people working, starting with the earliest hot days of May and then continuing progressively through into the terrific climatic heat, for example, in a foundry in July, then going on into September, it seems that what happens to the individual is not measured by temperature. Other things must happen, because some people break down under that strain. Pulse rate may be a measurement, but I think there are other things. In future tests to determine what high temperatures or humidities people can stand, we should find some better measurement than simply temperature rise of the individual.

F. W. HUTCHINSON, West Lafayette, Ind.: The comment I wanted to make is equally applicable to both the paper preceding [2] and this one with respect to the effective temperature. Dr. Winslow has already suggested that the operative temperature might be used more than the effective. I want to suggest the same thing and call attention to what seems to me a possible point of dispute. Particularly in the paper preceding this one the effective temperature was used as though it could be considered a line correlating work; the correlating effective ability to perform particular tasks. The effective temperature, if I am correct, was intended primarily as a heat balance correlation, and consequently I question whether there is any reason to believe that the 80 deg ET line means anything except at the particular point of dry bulb and wet bulb, for which the data were obtained.

CYRIL TASKER, Cleveland, Ohio: I am very glad that these discussions have come out and particularly that the last three speakers have raised points which are of very great importance to the Research Laboratory. It would seem that a great number of people in this Society have forgotten that the Effective Temperature Lines are lines of equal warmth. I am surprised sometimes to find how many people forget what is the definition of Effective Temperature. It is a line of equal warmth, and was established under conditions where the surrounding surfaces were all pretty much at the ambient temperature. Before I came to my present position I was one of the greatest critics of the idea that you could take these Effective Temperature Charts and apply them to conditions where the temperature of the surrounding surfaces varied very greatly, or to any serious degree, from the ambient temperature.

In our new Laboratory we intend to construct two psychrometric rooms side by side with easy access from one to the other, and one of the problems which we have on our schedule is a closer study of this effect of MRT or radiating walls, floor and ceiling on the lines of equal comfort, because our chart of Effective Temperature, while used widely, has been subject to a great deal of criticism on that very point.

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[2] Effect of Branch Take-Off Design on Noise in Ventilating Duct Systems, by Oscar Imalis, F. C. Houghten and Clark M. Humphreys (see Chapter No. 1289).

I hope those of you who have any other comments and criticisms to make, or from your experience think that there are points that deserve study, will bring them to the attention of the Laboratory. We cannot do anything more important in the next few months than to try to settle some of these points which are under discussion and about which there is a wide diversity of opinion. If radiant heating is to be the coming thing, then this Society should help develop some practical, factual data that its members can use, and this effect of radiation on our comfort lines, I think, is one of the most important things we have before us.

DR. WINSLOW: I am very glad Mr. Tasker brought that out. We do need that comfort effect but any analysis of figures with that in view should take into account what the actual heat demand of a given environment is. We have formulas well worked out and thoroughly established from which the air temperature, air movement, humidity and MRT being known, you can tell exactly what the heat demand of the environment is on the human body. However, you cannot measure this heat demand by determining the rectal temperature because of the fact that the body is adaptable to those extreme conditions and prevents the temperature from going higher. I do not think such results have much significance unless all the factors are known, including the sweat loss from the individual during the course of the experiment and particularly including the air movement and combination of those things in the formulas which are pretty well established.

MR. HUMPHREYS: I am not sure that I understand Mr. Fleisher's point. All our charts are based not on rise in temperature of the individual but on his final temperature.

MR. FLEISHER: I say they are not.

MR. HUMPHREYS: But they are. The charts in this paper do not indicate rise in temperature of the subject, but rather, his final rectal temperature. Mr. Avery pointed out that physical failures which cannot be accounted for by high body temperature, sometimes occur in industry. This is certainly true. A number of our subjects were removed, not because of high temperatures or high pulses, but because of what we have termed exhaustion. Neither their pulses nor their temperatures rose to the point that would call for their removal but they were just entirely unable to remain in the test.

Dr. Winslow stated that we should know something about sweat loss. That is known for these tests and is in the paper but for lack of time was not mentioned in this presentation.

I believe I have answered all the questions that were raised. There is just one comment I would like to make here in closing, and that is that this paper should not be taken as an indication that it is easy to get a 40-deg MRT elevation in your residence, if you so wish, with radiant heating. We do not know how much refrigeration was required to depress the dry bulb temperature sufficiently in the small test room to enable us to maintain an MRT elevation of as much as 40 deg, but we do know that one 7½-ton compressor was not sufficient and we had to install a second.

1291

No. 1291

AN EXPERIMENTAL INVESTIGATION OF THE EFFECT OF CHANGE IN ATMOSPHERIC CONDITIONS AND NOISE UPON PERFORMANCE †

By MORRIS S. VITELES,* PH.D., PHILADELPHIA, PA., AND KINSLEY R. SMITH,** PH.D., STATE COLLEGE, PA.

A study conducted under the auspices of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

INTRODUCTION

IT IS GENERALLY recognized that major problems arise in providing and maintaining comfortable atmospheric conditions for personnel working in confined spaces on Naval ships, particularly when the latter are cruising in tropical waters. Any attempt to improve atmospheric conditions either by using larger ventilating fans or through mechanical refrigeration immediately involves an increase in the weight of equipment. Weight considerations are extremely important in the design of Naval ships, and an increase in the weight of equipment can be justified only if it will improve the overall performance of the ship as a fighting unit. Moreover, the functioning of ventilating or refrigeration equipment is frequently accompanied by an increase in noise level which may in itself be detrimental to the performance of ship personnel.

Pertinent in this connection is the effect of hot atmospheres and higher noise levels on men in plotting and charting rooms, where a great deal depends upon the speed and accuracy of work and on the morale of personnel engaged in such work. The purpose of the investigation described in this report was

† This paper presents a detailed summary of a comprehensive report which could not be published in these Transactions because of space limitations. A limited number of copies of the complete report are available and application should be made to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

* Professor of Psychology, University of Pennsylvania.
** Assistant Professor of Psychology, Pennsylvania State College.

Presented at the 52nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January 1946.

to determine the effect of diverse atmospheric and noise conditions upon output, feelings, and physiological functions during the performance of tasks roughly analogous to operations carried on by personnel in the plotting and charting rooms of Naval vessels. The investigation was designed to determine whether the accuracy, speed, and attitudes of personnel could be improved by air conditioning and the extent to which such improvement through reductions of temperature might be counter-balanced by possible adverse effects of increased noise.

ACKNOWLEDGMENTS

The general aspects and requirements of the study were outlined by Mr. A. E. Stacey, Chairman of the Committee on Research, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; Mr. J. H. Walker, Technical Adviser; the late Dr. F. C. Houghten, Director of Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; and Comdr. R. M. Watt, Jr., and members of his staff and associates in the U. S. Navy. The experiment was designed and conducted by the authors in consultation with the Committee on Research and Navy personnel and, most particularly, with the assistance of the late Dr. F. C. Houghten, who was responsible for the choice and administration of the physiological measures employed in the study.

The study was carried out in 1940 in the psychrometric rooms of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, then located in the U. S. Bureau of Mines Building, Pittsburgh, Pa. S. C. Markey, H. T. Olsen, Carl Gutberlet, A. A. Rosenburg, and John Suciu, Jr., of the laboratory staff were largely responsible for maintaining the physical conditions of the study and for collecting the data. They also cooperated in the preliminary analysis of the data, under the direction of the authors, who were responsible for the statistical analysis and who prepared the final report which is summarized in this article.

GENERAL CONDITIONS OF THE MAIN EXPERIMENT

The main experiment involved an analysis of changes in *output, feelings,* and *physiological* state of six subjects [1] who worked on selected experimental tasks six days per week for a period of seven weeks.

Subjects

The subjects were selected from a group of 40 candidates through application of standards designed to provide, (1) an experimental group meeting the physical qualifications for acceptance by the Navy, and (2) a group relatively homogeneous from the viewpoint of age, education, general intelligence, and initial scores on the experimental tasks employed in the main experiment. The general characteristics of the group of subjects used in

[1] The investigation was initiated with eight subjects. Two subjects dropped out during the fourth and fifth weeks. Although replacements were obtained for these subjects, in order that social factors would be held as constant as possible, only data from the six subjects who remained throughout the experiment were considered in the analysis.

the investigation were as follows:

1. *Physical Conditions:* Passed physical examination for United States Navy. Diagnosed as essentially normal in hearing on basis of complete audiometric examination administered through the co-operation of the Pittsburgh Public School System.

2. *Weight:* From 129¼ lb to 168½ lb.

3. *Height:* From 5 ft 7 in. to 6 ft 2 in.

4. *Age:* From 18 to 21 years.

5. *Education:* High school graduates with no further formal education.

6. *Mental Alertness:* Subjects ranged between *average* and *above average* in general intelligence as measured by the *Otis Self-Administering Test of Mental Ability* (Higher Form). Raw scores ranged from 43 to 55, giving an I.Q. range of from 101 to 113, *i.e.,* within approximately +1 standard deviation from the average.

7. *Initial Performance on Experimental Tasks:* The tests, or experimental tasks, used in the investigation were administered, and only those applicants were selected who showed no wide deviation from average performance on these tests (as determined from available norms).

Atmospheric Conditions

Atmospheric conditions included effective temperatures of 73 deg (79 F dry bulb—65.5 F wet bulb); 80 deg (88 F dry bulb—73.3 F wet bulb); 87 deg (98 F dry bulb—81.5 F wet bulb), and 94 deg (108.5 dry bulb—90.3 F wet bulb).[2] These temperatures were maintained through the operation of the temperature control room available in the Research Laboratory of the A.S.H.V.E.

Noise Levels

Noise levels of 72, 80, and 90 db (decibels) were employed during the main experiment. The source of noise was a fan, installed and operated under conditions specified by U. S. Navy personnel. Electrical transmission was employed in bringing the noise into the experimental room.

Experimental Tasks

The Discrimeter: This task consisted of a large upright disc about 18 in. in diameter, near the top of which was a small window through which a single digit was visible. Below the disc were four keys, one for each finger of the subject. The subject kept the fingers of his right hand upon the keys and pressed the finger corresponding to the digit which was visible in the window. If the wrong key was pressed the mechanism did not respond. Pressing the right key brought into view another digit which was responded to in turn. The score was expressed as the number of correct responses made in 30 min, and was automatically recorded by an electric counter.

The Mental Multiplication Test: This test presented a number of multiplication problems calling for the product of a three-digit and a two-digit

[2] *Plotting Room* conditions were given major consideration in the choicest of atmospheric and noise conditions to be investigated.

number. Performance time was one-half hour, the score being determined by the number of correct digits in the answers. An error score could also be computed in terms of the number of incorrect digits in the answers.

The Number Checking Test: This task involved inspecting pairs of numerical series and checking those pairs in which both numbers of the pair were identical.

The Lathe Test: This apparatus duplicated the slide rest of an engine lathe by means of which the cutting tool is controlled. In place of the cutting tool there was an electrified stylus or pointer. The subject traced the stylus in a horizontal plane around a circular pattern by turning the two control handles. Tracing off the pattern counted as an error. Tracing correctly one-sixth of the circle constituted a correct response. Errors and correct responses were recorded automatically by electric counters. Performance time was one hour.

The Typewriter Code Test: A modified typewriter having a shield between its roller and the subject was used. Through this shield could be seen at one time only one letter of a row of letters on a sheet inserted into the typewriter in the ordinary manner. Ten different letters were used and were presented in scrambled order in long rows. For each letter an appropriate key of the ten keys in the top row of typewriter keys was to be pressed. These keys were not the same as the letters, but represented a coded response, one symbol for each letter. The response was made right on the stimulus sheet in the typewriter in the form of a typed symbol. Performance time was one-half hour.

The Locations Test: On each page of this test were large squares containing letters arranged in definite positions in rows and in columns. Around each of these squares were smaller squares or *blocks,* in which dots were located in the same relative position in the small square as was a certain letter in the larger square. The subject wrote over each dot the appropriate letter indicating the corresponding position of the letter in the larger block. Dots correctly located represented the score, incorrect locations represented the error score. Fifteen sheets constituted a test form. Performance time was one-half hour. (This test was an adaptation of a part of the MacQuarrie Test for Mechanical Ability.)

The Pursuit Test: On each page of this test was a number of mazes each consisting of intertwined lines. In each maze 10 lines started at numbered blocks on the left. Each line was to be traced by eye to the block at the right in which it ended. The number of the starting block was to be placed in the block at which the line terminated. The score was represented by the number of lines correctly followed and recorded. Lines incorrectly followed and incorrectly recorded were scored as errors. Performance time was one-half hour. (This task was also an adaptation of part of the MacQuarrie Test for Mechanical Ability.)

Physiological Tests: Physiological tests included pulse rate and oral temperature taken at stated intervals.

Feelings and Attitudes: Changes in attitudes and feelings of subjects during the progress of the study were investigated by means of a questionnaire in terms of which responses of the subjects were obtained on pertinent questions.

PROCEDURES IN THE MAIN EXPERIMENT

Administration of Tests: The main experiment consisted of 42 experimental sessions, each 5½ hours in length, held 6 days a week for a period of 7 weeks.

Pre-test Practice: In order to familiarize subjects with the experimental tasks and to provide an opportunity for concentrated learning prior to the initiation of the main experiment, a period of practice was introduced before the beginning of the experiment proper. The practice period consisted of 4 sessions, each 2 hours in length, during which each subject worked on all of the experimental tasks at a noise level of 72 db and with effective temperature conditions of 73, 73, 80 and 87 deg ET, respectively, on the four consecutive days. (See Table 1.)

Experimental Sessions: Experimental tasks were administered on the basis of carefully standardized procedures designed to equalize the influence of practice and of variations in output related to hour and day of work. The order of tests was varied for each subject from week to week, but during the course of the experimental series all subjects were given the same tests in the same order with approximately the same frequency. In addition, in the case of the pencil-and-paper tests, enough alternative test forms were provided so that no subject was given the same blanks on two successive days.[3]

The procedure for the experimental sessions was as follows:

1. Each subject reported to the laboratory 45 min before the experimental session and was provided with a standard lunch.

2. Each subject then spent 45 min at rest in an unconditioned anteroom with temperature at that prevailing in the building. The last few minutes of this 45-min period were spent in obtaining readings on pulse and oral temperature.

3. The subjects then entered the experimental room and immediately started working at the experimental tasks under temperature and noise conditions applying for the experimental session. Work on the experimental tasks in this portion of the experimental session was continued for a period of 2 hours.[4] At half-hour intervals during this work period each subject's pulse rate and oral temperature were taken by an experimenter without stoppage of work on the part of the subject. Pulse rate was taken at the temple, and temperature through insertion of a thermometer in the mouth of the subject. These measures were always taken within the last 5 min of the working spell.

4. Immediately upon the completion of this portion of the work session, subjects were allowed a 5-min rest period in the experimental room. During this time physiological measurements were repeated with the subject at rest.

5. Following the rest period subjects again spent 2 hours at work on the experimental tasks in accordance with the standard pattern. As in the first portion of the work period, pulse rate and oral temperature readings were taken at 30-min intervals.

6. At the end of the second work period subjects proceeded to the anteroom and spent a minimum of 30 min at rest before dismissal. Pulse rate and oral temperature readings were taken towards the close of this rest period.

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[3] With the exception of the Locations Test.
[4] One-minute intervals were allowed between 30-min work periods for change-over from task to task.

TABLE 1—ORDER AND NATURE OF THE EXPERIMENTAL PERIODS (1940)

MARCH 4	MARCH 5–8	MARCH 9	MARCH 11–30	APRIL 1	APRIL 2–6	APRIL 8–20	APRIL 22–27	APRIL 29	APRIL 30 TO MAY 5
Various tests administered in selection of subjects	Pre-Practice Periods Noise—69–74 db ET as follows: Date — Deg ET 5 — 73 6 — 73 7 — 80 8 — 87	Speed, Trial Period 72 deg ET Noise—73 db	Series II Three Weeks of Experimental Work Periods Noise constant—72 db ET systematically varied daily thus: Date — Deg ET 11 — 87 12 — 80 13 — 73 14 — 73 15 — 80 16 — 87 18 — 80 19 — 73 20 — 87 21 — 87 22 — 73 23 — 80 25 — 73 26 — 87 27 — 80 28 — 80 29 — 87 30 — 73	Speed, Trial Period 72 deg ET Noise—73 db	Intermission	Series III Two Weeks of Experimental Work Periods Group reduced from 8 to 6 subjects beginning with this Series. Two new subjects added to keep size of group constant but work of the two new members not averaged in with the original six. Effective Temperature and noise level varied daily systematically thus: Date — Deg ET — Noise Level db 8 — 73 — 80 9 — 87 — 90 10 — 80 — 80 11 — 80 — 90 12 — 87 — 80 13 — 73 — 90 15 — 73 — 90 16 — 87 — 80 17 — 80 — 90 18 — 80 — 80 19 — 87 — 90 20 — 73 — 80	Series IV Control Period One Week—same as third week of Series II	Speed, Trial Period	Series V Six Days of Experimental Work Periods Noise constant at 72 db ET systematically varied daily, thus: Date — Deg ET 30 — 80 1 — 94 2 — 87 3 — 80 4 — 94 5 — 87* (*Unable to maintain temperature on this day)

Motivation of Subjects: Subjects were paid a base rate of 40 cents per hour for all time spent on the experiment, exclusive of lunch periods. In addition, bonus payments were made to subjects on both a group and individual basis to encourage maximum effort on the part of each subject throughout the experiment.

Attitude Questionnaire: One purpose of the study was to examine changes in attitude and feelings occurring under various conditions of noise and temperature. These problems were investigated by means of two questionnaires and in terms of a written statement by each subject. The first of these questionnaires referred to attitudes and feelings resulting from temperature conditions and was filled out by each subject at the close of the final session of Experimental Series II. The second questionnaire, dealing with reactions to noise conditions, was administered approximately one week after the completion of Experimental Series IV. Written statements were obtained from the subjects following their first work period at 94 deg ET in Experimental Series V.

CRITERION MEASURES

Production Data: Accurate records were kept of production during successive five-minute intervals on each test by means of Veeder counters and electric timers in the case of the apparatus tasks, and by requiring subjects to indicate by check marks the amount of work completed in successive 5-min periods in the case of paper-and-pencil tests. From these records the following indices relating to production were computed:

1. *Output.* This measure referred to the total number of work units performed without respect to errors.

2. *Error Ratios.* This measure was computed for each individual on each test (except the Discrimeter) by dividing total number of errors by total output including errors. Error scores on the Discrimeter were not available.

3. *Variation Percentages.* As indicated previously, records of output were obtained at 5-min intervals on all tests. Differences in output between successive 5-min periods of work on each test by each subject were averaged and then divided by the subject's average 5-min output on the task to obtain the variation percentage for the subject. This ratio gave an indication of fluctuation in output during successive 5-min intervals of the work period.

Physiological Measures

1. *Oral Temperature.* Oral temperature was taken in tenths of degrees Fahrenheit. Measurements for each subject were expressed in terms of increase or decrease in oral temperature during the period of work in the experimental room as compared with the oral temperature readings taken during the rest period immediately preceding the work period.

2. *Pulse Rate.* Pulse rates were recorded in terms of beats per min and were expressed as increase or decrease of pulse rate during the experimental period over that obtained during the rest period immediately preceding the work period.

Questionnaire on Feelings and Attitudes: No attempt was made to provide for quantitative treatment of data from these questionnaires.

RESULTS

The data from this experiment were treated graphically, and the significance of differences between performances of the men under the several experimental conditions evaluated statistically by means of the t test.[5] Due to the limitations of space all of the graphs and tables cannot be presented in this report. However, the results of the complete analysis have been taken into considera-

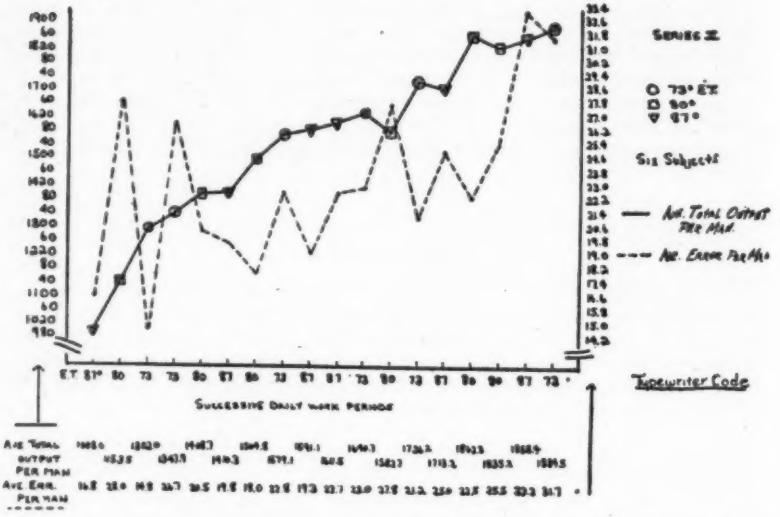

FIG. 1. RELATION BETWEEN TIME AND THE AVERAGE OUTPUT AND ERROR PER MAN FOR TYPEWRITER CODE

tion in the evaluation of the results which follow, and illustrative tables will be presented.

Effect of Practice: In the case of every test improvement was demonstrated with increased practice. In other words, learning persisted throughout the

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[5] The significance of observed differences has been statistically determined with use of the following formula for the standard error of the difference between means in terms of differences between paired scores:

$$\sigma(ms - my) = \frac{\sigma(x - y)}{\sqrt{N - 1}}$$

In each case, Fisher's technique for use in treatment of small group data has been applied in interpreting the relationship between the difference between means and the standard error of that difference. This technique provides a P-value which indicates the probability that an observed difference in two performances is due to chance. A P-value of 0.01 means that the probability that an observed difference is due to chance is only 1 in 100. Where the P-value is between 0.02 and 0.05 the probability is from 2 to 5 in 100 that the observed difference between means is due to chance. P-values of 0.05 and under are generally regarded as significant, values of 0.01 as very significant. . . . (R. A. Fisher, *Statistical Methods for Research Workers*, 5th Ed., Edinburgh, Oliver and Boyd, 1934, p. 118.)

entire experiment. This trend is evident, for example, in Fig. 1 in which are graphed the average total output per man, and the average errors per man in the various experimental sessions throughout the experiment in terms of the Typewriter Code Test. However, the influence of such learning so far as analysis of results is concerned is made constant by the fact that the average position and deviation (in rank) of experimental trials is identical for each of the temperature conditions, 73, 80 and 87 deg, respectively, and for each of the noise levels, 72, 80, and 90 db, respectively.

Effect of Changes in Effective Atmospheric Conditions

Output: Output on the varied tasks used in this investigation is not adversely affected by a change from 73 to 80 deg ET. As a matter of fact, maximum

TABLE 2—GROUP TOTAL PERFORMANCE OUTPUT AT VARIOUS EFFECTIVE TEMPERATURE LEVELS

SERIES II

(*Six Subjects Who Finished All Series*)

PERFORMANCE	GROUP OUTPUT (%)			SIGNIFICANCE OF DIFFERENCES (P)		
	73 ET	80 ET	87 ET	73–87 ET	80–87 ET	73–80 ET
Mental Mult.....	100.0	99.1	96.7	0.01–0.02	0.02–0.05	0.40–0.50
No. Checking.....	100.0	99.6	95.0	0.02–0.05	0.02–0.05	0.70–0.80
Lathe.....	98.5	100.0	86.1	0.02–0.05	0.01–0.02	0.20–0.30
Type. Code.....	100.0	98.1	96.5	0.20–0.30	0.40–0.50	0.20–0.30
Discrimeter.....	100.0	98.0	97.3	0.20–0.30	0.70–0.80	0.20–0.30
Locations.....	99.0	100.0	97.2	0.10–0.20	0.05–0.10	0.30–0.40
Pursuit.....	98.1	100.0	96.5	0.10–0.20	0.01–0.02	0.05–0.10

output is obtained almost as frequently at 80 deg ET as at 73 deg ET. The differences in performance on individual tasks when these two effective temperatures are compared are small in extent and are statistically insignificant in every instance. This is evidenced by reference to Table 2 in which the output data are summarized from experiments in Series II, where Effective Temperature was varied with the noise level held constant at 72 decibels.

However, at an 87 deg ET output is consistently lower than at 73, and 80 deg ET, respectively. As is indicated by Table 2, the reduction in output ranges from 2.7 to 13.9 per cent of maximum output, but only in two cases (Lathe Test and Number Checking) is the reduction as great as 5 per cent. Differences between output at an 87 deg ET and at the lower effective temperatures are found to be statistically significant at the 5 per cent level or lower in the case of four tests, namely: Mental Multiplication, Number Checking, Lathe, and Pursuit Test.

Accuracy: Except in the case of the Lathe Test the percentage of errors in response remains practically unchanged with variations in the range of 73 to 87 deg ET. Even in the case of the Lathe Test the absolute increment in error ratio is not great. Moreover, neither in the case of the Lathe Test,

nor of any other test, were changes in accuracy accompanying changes in temperature statistically significant.

Variability of Performance: Analysis of the variation percentages indicated that performance at an 87 deg ET tended to be more variable than performance at the lower effective temperatures. In other words, there was a tendency for greater fluctuation in the amount of work done in successive 5-min periods at 87 deg ET than at 73 and 80 deg ET, respectively. However, the amount of absolute difference is not great in the case of any of the tasks employed and is statistically significant only in the case of the Number Checking and Discrimeter Tests.

Physiological Measures: In addition to the production measures, it is necessary to consider the effect of the experimental temperature conditions on physiological changes. The data indicated that there was a consistent increase in both pulse rate and oral temperatures with increase in effective temperature from 73 deg through 80 to 87 deg, and that the differences in both pulse rate and oral temperature from 73 deg through 80 to 87 deg, respectively, are statistically significant. The amount of increase in pulse rate and oral temperature (over readings during rest) is most marked at the 87 deg level, the average increase at that level being approximately 1½ F for oral temperature, and approximately 20 beats per minute for pulse rate.

General Conclusions with Respect to Changes in Effective Temperature: On the basis of these findings it may be safely said that for work periods not exceeding 4 hours on tasks of the type employed in this investigation no adverse effects on performance are produced by an increase from 73 to 80 deg ET. With an increase to 87 deg appear indications of adverse effects upon quantity and variability of performance. While these are not marked, the indications from the main experiment, combined with those of the supplementary experiment (Series V) described below, suggest that as effective temperature is increased to 87 deg, there is an approach to a *danger zone* so far as the effect of temperature on performance is concerned.

TABLE 3—TOTAL PERFORMANCE OUTPUT AT VARIOUS NOISE LEVELS

SERIES II (LAST WEEK) III & IV

(*Six Subjects*)

PERFORMANCE	GROUP OUTPUT (%)			SIGNIFICANCE OF DIFFERENCES (P)		
	72 db	80 db	90 db	72–90 db	80–90 db	72–80 db
Mental Mult.....	94.8	96.5	100.0	0.01	0.01–0.02	0.30–0.40
No. Checking.....	94.1	98.9	100.0	0.02–0.05	0.20–0.30	0.05–0.10
Lathe.....	100.0	97.4	97.6	0.40–0.50	0.90–1.00	0.20–0.30
Type. Code.....	97.9	100.0	99.7	0.40–0.50	0.80–0.90	0.30–0.40
Discrimeter.....	100.0	95.6	97.3	0.20–0.30	0.50–0.60	0.05–0.10
Locations.....	99.8	99.0	100.0	0.70–0.80	0.50–0.60	0.60–0.70
Pursuit.....	97.8	98.7	100.0	0.30–0.40	0.20–0.30	0.60–0.70

The Effect of Changes in Noise Conditions

Output: Analysis of the data indicated that increase in noise level above 72 db, considered apart from the effect of temperature, appears to be somewhat favorable to increased output.[6] This is evident, for example, from examination of Table 3. The effect is most marked at the 90 db level which furnishes maximum output in the case of 4 of the 7 tasks, including all of the paper and pencil tests involving a minimum of physical movement and a maximum of intellectual activity. Output is highest at the 72 db level only in the case of two tasks (Lathe Test and Discrimeter). However, the amount of superiority of output at the 90 db level is in no instance great and the difference between output at this level of noise and at that level which gives minimum output is statistically significant at the 5 per cent level or lower only in the case of Mental Multiplication and Number Checking tasks. In the latter case a statistically significant difference is evident only in the comparison of performance at 72 and 90 db, respectively.

Accuracy: Accuracy was not consistently or seriously affected by increase in the amount of noise from 72 to 90 db. In the case of two tasks, Mental Multiplication and Lathe, there was a statistically significant adverse effect upon accuracy with increase of noise to 90 db, but even in the case of these tests the total loss of accuracy is relatively negligible, the error ratios under the 80 and 90 db conditions being 2.98 and 3.23, respectively, in the case of the Mental Multiplication, and 16.54 and 17.23, respectively, in the case of the Lathe Test. The Locations Test actually shows a statistically significant decrement in error ratio at the 90 db level.

Variability of Performance: Variability of performance is not affected markedly by changes in the amount of noise. On some tasks the variability of performance increases slightly with increasing noise, on other tasks, the variability of performance decreases slightly with decreasing noise. Only in the case of the Pursuit Test was there a statistically significant increase in variability with increase in noise, the total amount of change again being negligible, the variation percentage increasing from 12.39 to 14.31 in this case.

Physiological Measures: Examination of the data indicated no marked changes in terms of pulse rate and oral temperature associated with changes in noise level. In only four cases were differences statistically significant at the 5 per cent level or lower. Three of these arose in comparisons of oral temperature at the 72 and 80 db levels.

General Conclusions with Respect to Changes in Noise Levels: The findings of this experiment support the conclusion that performance on tasks such as those used in this experiment is not adversely affected by a rise in noise level to 90 db. On the basis of psychological data alone it would seem that expenditure of funds for the reduction of noise as a means of improving quality of performance during a 4-hour period of work on the type of tasks involved in this investigation is not warranted.

[6] It is interesting to note that these results on increased output with increasing noise are in conformity with other experiments of the same kind. However, these other experiments also show an increase in energy output in noisy as compared with less noisy conditions.

Effect of Combined Changes in Atmospheric and Noise Conditions: Production under stated combinations of noise and atmospheric conditions is presented in Table 4. Examination of this table indicates that in the case of every task but the Discrimeter, the lowest output occurred with conditions at the 87 deg ET level, the lowest output for the latter task occurring at the 80 deg ET. Even in this case this level of output is very little below that obtained at the 87 deg level with noise at 72 db. A similar tendency is observable with respect to errors. On the other hand, when atmospheric and noise conditions are considered in combination there is no consistent tendency for lowest output to be associated with any one of the three noise levels. Actually, in two tasks lowest output occurs at the 72 db level, in two other tasks at the 80 db level, and in the case of the remaining three tasks at the 90 db level.

Results of Supplementary Experiment (Series V). As noted previously in this experiment it was planned to compare the influences of 80, 87 and 94 deg

TABLE 4—TOTAL PERFORMANCE OUTPUT AND RATIOS OF ERRORS TO TOTAL PERFORMANCE OUTPUT UNDER VARIOUS COMBINED EFFECTIVE TEMPERATURE-NOISE LEVEL CONDITIONS[a]

SERIES II (LAST WEEK) III & IV

(*Six Subjects*)

DEG ET	72 db		80 db		90 db	
Mental Mult.						
73	89.5	(2.86)	88.6	(3.18)	96.8	(3.20)
80	90.9	(2.70)	100.0	(2.65)	94.2	(3.38)
87	91.4	(2.90)	**88.2**	(3.11)	95.7	(3.20)
No. Check						
73	89.1	(2.78)	92.0	(2.78)	100.0	(3.22)
80	90.3	(3.19)	95.0	(2.60)	97.7	(3.41)
87	**88.2**	(2.86)	90.8	(3.79)	90.7	(3.21)
Lathe						
73	100.0	(10.3)	94.9	(16.3)	95.2	(15.1)
80	95.8	(12.8)	91.5	(15.7)	98.2	(16.2)
87	84.4	(12.7)	**86.4**	(16.3)	**80.0**	(19.4)
Type. Code						
73	95.4	(1.97)	99.3	(2.09)	98.8	(1.81)
80	97.2	(2.04)	99.8	(2.26)	100.0	(2.31)
87	95.3	(2.05)	94.8	(2.22)	**94.3**	(2.07)
Discrimeter						
73	100.0		93.9		89.2	
80	95.9		86.4		97.2	
87	87.1		90.4		89.1	
Locations						
73	96.2	(1.06)	96.7	(1.18)	94.7	(1.09)
80	96.9	(1.15)	93.8	(1.18)	100.0	(1.08)
87	93.5	(1.32)	93.9	(1.28)	**92.6**	(0.95)
Pursuit						
73	95.8	(0.85)	94.7	(0.82)	97.4	(0.84)
80	97.6	(0.78)	94.4	(0.82)	100.0	(0.65)
87	**92.7**	(0.93)	97.3	(0.89)	95.1	(0.99)

[a] Total output is expressed as a percentage of the maximum appearing under any of the conditions. Error ratios appear in parentheses.

ET, respectively, with noise held constant at 72 db. It was originally planned to obtain two days of experimental readings under each temperature condition with each subject. Because of interruption in heating service, it was possible to obtain only one experimental day of readings with an 87 deg ET.

Because of this fact, and since none of the subjects were able to complete the tasks upon their first exposure to a 94 deg ET, the volume of data was insufficient for the statistical analysis of data on this condition. However, it is significant to note that none of the subjects were able to complete their first four hour period at 94 deg ET and all of them reported physiological effects such as dizziness, partial or complete visual blackout, and nausea at this effective temperature. There was one case of nose-bleed and some staggering or swaying on the part of almost all of the subjects. There were marked increases in both oral temperature and pulse rate. At 94 deg ET the average increase in oral temperature was approximately 2½ F, and the average increase in pulse rate approximately 45 beats per minute.

Under the 94 deg ET conditions output was considerably reduced and error scores considerably increased for most of the tests. The reduction in output as compared with that under an 80 deg ET varied from 14 to 42.6 per cent for the various tasks. Increase in variability also was noted. The results of this supplementary experiment again suggest that there is an approach to a *danger zone* as temperature is increased to 87 deg and that an increase to 94 deg is definitely and highly detrimental to performance. It is apparent that further investigations should concern themselves with the influence of effective temperatures between approximately 80 deg and 90 deg.

Changes in Feelings and Attitudes Associated with Various Noise Levels and Temperature Conditions: The principal source of data on feelings and attitudes of the subjects during the experiment was a questionnaire presented to the subjects at the close of the main and of the supplementary experiment respectively. It was not possible to treat these data quantitatively. However, following is a summary of the major findings obtained from an inspection of questionnaire data:

1. Subjects were not able to identify with any degree of accuracy the nature or range of variations of both temperature and noise employed in this experiment.

2. Under conditions of 94 deg ET subjects definitely report increased discomfort and feelings of annoyance and distress.

3. There is no indication of marked discomfort or annoyance and distress at 73 deg or 80 deg ET. At 87 deg ET all of the subjects reported unfavorable reactions such as feelings of discomfort, sluggishness, irritability, and fatigue.

4. The reports on the noise questionnaire indicated a very rapid adjustment to increased noise with no indication that even a noise level of 90 db exercises a seriously disturbing influence on attitude of subjects under the conditions of this investigation.

SUMMARY AND CONCLUSIONS

The following tentative conclusions may be drawn from a consideration of the results:

1. The findings suggest that there is no need to provide equipment for the reduction of effective temperatures below the 80 deg level where tasks of the type included

in this investigation are to be performed during work periods not exceeding 4 hours. There is some evidence that it may be undesirable to do this type of work at effective temperatures as high as 87 deg but further investigation, within the range of approximately 80 to 90 deg is required to determine more exactly the critical level of temperature.

2. The evidence suggests that it is unnecessary to undertake considerable expenditure of funds to reduce the noise involved in the cooling and circulation of air, as a means of improving quality of performance during a 4-hour work period on the type of tasks involved in this investigation, at least if such noise does not rise above the 90 db level. In general it would appear that the effect of high temperature on performance is much more adverse than that of high level of noises of the type reproduced in this experiment.

DISCUSSION

B. M. BROD, La Guardia Field, N. Y.: What was the frequency of the noise imposed on the subjects? Did this or any other test indicate that the frequency of the noise imposed resulted in a physiological or psychological effect? In other words, is high frequency noise more or less annoying than low frequency noise?

DR. VITELES: I wish I could answer both of these questions. I can answer neither because we depended upon the Navy to set up the noise conditions. The Navy mixed the noise for us and presumably used the type of noise found on naval vessels in terms of the frequencies involved. We, therefore, had no opportunity in this experiment to compare the effect of high frequencies with those of low frequencies. I cannot think of any experimental study bearing upon this question at the moment, however, there may be some.

This whole matter of the effect of noise becomes complicated. Generally speaking we find in experiments that have been conducted elswhere that noise does not adversely affect work output, but experiments in which carbon dioxide output or other measures of physiological fatigue have been used, have shown that there is a tendency for subjects to use more effort in producing the same results under high noise conditions. That did not appear in this experiment, and, quite frankly, I am not sure this was not due to the nature of the measures that were used. These measures were selected by the late Dr. Houghten.

W. L. FLEISHER, New York, N. Y.: Of course we are all very much interested in this subject because it is on this basic subject that we really started the Research Laboratory. I am not going to discuss that particular point. The thing I am interested in and have been right along on the question of Effective Temperatures, is the relationship between the dry bulb temperature and the wet bulb temperature, which makes, of course, Effective Temperature. With the 80-deg ET, I am very much interested in finding out the dry and wet bulb temperatures and have always felt that we should not use effective temperatures in these test cases. There may be a great variation in the relationship of the sensible and latent heat losses. So I believe that this should be separated or segregated, and I would ask if anyone has information as to whether the 80-deg ET under which everyone seems to be able to perform with the same ability as at the 75 deg ET, was the dry bulb at the 90 or 95 deg sensible temperature level, or whether it was closer to the 80 deg dry bulb temperature.

Of course the importance of this would readily be understood because it would affect the nature of the design of a system which could produce these results. If anyone has that information, I would like to have it.

C. S. LEOPOLD, Philadelphia, Pa.: The speaker was very careful to point out that these experiments were made with extremely high incentive, and I just have a word of caution in applying the reasoning to the average and low incentive jobs.

It is not comparable, and I wish in summary, Dr. Viteles, you could give us a bit of thought or observation on that point. I think it would be quite helpful.

J. P. BAND, Ottawa, Canada: Would there be any information on results of the tests if the men were working under the same conditions say for two months prior to taking the tests? Acclimatization has quite a distinct effect, particularly on work in hot climates.

R. F. REAS, Chicago, Ill.: I wonder what the effect would be on the subjects as they continue for six weeks as to increasing proficiency? Were you able to correlate results at the end of the test period with those in the early part as the subjects became more proficient in the tasks they were doing?

DR. VITELES: So far as the Effective Temperatures are concerned, 73 deg ET was 79 deg dry bulb and 65.5 deg wet bulb. The 80 deg ET was 88 deg dry bulb and 73.3 wet bulb. The 87 was 98 dry bulb, 81.5 wet bulb, and the 94 was 108.5 dry bulb and 90.3 wet bulb.

Mr. Fleisher has brought up a very important point. In a complete investigation of this area it is necessary to take the entire range of humidities, and the entire range of conditions which produce an Effective Temperature at a particular time. Effective Temperatures used in the experiment were selected I think by the Navy. or perhaps by the Navy and the Society, as representing, I assume, the average of overall conditions.

I am very glad that this matter of motivation was raised because if I were in a position to plan such an experiment again for the Navy, I would most certainly not be inclined to use the kind of motivation that was employed in this experiment. I have learned a lot about the Navy since, and one of the things that I have learned is that the kind of motivation which we ordinarily know exists in industrial conditions, does not exist to anywhere near the same extent in the case of day to day operation of seamen or perhaps even rated personnel, in the Navy. I have a feeling that it was entirely unrealistic, in terms of the application of these results at least to the peace time Navy, to use such high motivation. I know a little more about the Navy now than I did in 1940 and I think that this knowledge would influence a repetition of this experiment.

In the second place, there was another element of this experiment which was unrealistic and which has been touched upon by Mr. Band, who asked whether a greater length of time for acclimatizing the subjects to the conditions would change the results. I have a feeling that seven weeks was probably long enough in acclimatizing the subjects but there was one weakness in this experiment. It was conducted during the late winter and early spring of 1940, in Pittsburgh. When the subjects left the room they went out into what was pretty comfortable weather and they slept, in the main, under pretty comfortable conditions. If I were to repeat this experiment I think I would be inclined to insist upon doing it in a naval center where we could get the subjects to live under the kind of conditions which apply in actual naval operations. Probably the best way to do it would be aboard ship. I think we have set a pattern in this war in the matter of conducting research aboard ships of which we might well take advantage in future research.

I may have a bias but I think I would be inclined also to introduce other measures of physiological change. This might bear discussion but from what I know about the use of these various measures I personally have a bias toward strong carbon dioxide output.

The question was asked about the comparison of the proficiency at the end of the experiment with proficiency at the beginning of experiment. Under ideal conditions we should use tasks which do not show the effect of improvement.

There is a committee working now, composed of Army and Navy personnel and civilians, trying to set up a standard battery of tasks which, among other things, will be characterized by lack of improvement as a result of practice in the tasks. It is my humble opinion that the Committee is facing an almost impossible job, and I question very much whether we shall ever reach the point of finding a task that is sufficiently complex to meet our needs and at the same time be entirely free from practice effect. So we are always faced with the problem of comparing results at the end with results at the beginning. We have made no such comparisons. We did not study these results in terms of change from week to week. However, we followed the best technique which we could, for making the practice effect constant for varying temperature conditions by making identical the average position and deviation (in rank) of experimental trials under each of the temperature conditions.

1292

No. 1292

THE SHIELDING OF THERMOCOUPLES FROM THE EFFECTS OF RADIATION

By GEORGE V. PARMELEE * AND RICHARD G. HUEBSCHER,** CLEVELAND, OHIO

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the Research Laboratory, Cleveland, Ohio.

PURPOSE

THIS REPORT presents the results of a study which was made to determine how effectively shields might reduce the effects of radiation on thermocouples used to measure room air temperatures. The need for such quantitative data arose during the construction at the A.S.H.V.E. Research Laboratory of apparatus designed to measure the transmission of solar heat through glass. It was desired to measure accurately the temperature of the air within a 25 cu ft space enclosed by blackened sheet metal and a panel of glass under test. During a test the thermocouples would be subjected not only to the radiating influence of the sun but also to that of the glass, the temperature of which might differ considerably from that of the enclosed air. It was anticipated that the temperature of the metal surfaces would be within a few degrees of that of the air.

No extensive literature search was made, although, undoubtedly, this problem has often been discussed before. Information found in a book on temperature and its measurement [1] dealt, for the most part, with the temperature measurement of hot gases flowing through flues, and suggested the use of aspirated and shielded couples of various types. In the apparatus mentioned simplicity of installation and avoidance of air movement, other than that created by natural means, were considered sufficiently important to warrant some method of shielding. It was believed that a more useful purpose would be served by presenting the results of this investigation at the present time rather than

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[1] Temperature, Its Measurement and Control in Science and Industry, Reinhold Publishing Co., New York, 1941.

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including them in a longer report at a later date. Although the scope of the investigation was limited, such data as were obtained established some quantitative relationships that may be of value to others meeting similar problems.

EXPERIMENTAL APPARATUS

Preliminary studies, made with direct sunlight indoors and with several different arrangements for shielding the thermocouples, led finally to the construction of the apparatus shown schematically in Fig. 1. In the tests herein discussed, the radiating source consisted of a cone type heater element operating at a red heat (wire at approximately 1600 F and ceramic at 1100 F). A cardboard enclosure surrounded the thermocouples under test and cardboard radiation shields were placed between the source and the enclosure. Therefore,

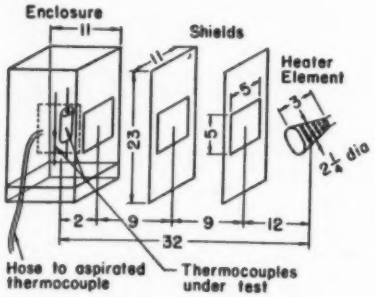

FIG. 1. APPARATUS FOR DETERMINING EFFECTIVENESS OF THERMOCOUPLE SHIELDS

with the exception of the small heater all surfaces seen by the thermocouples were substantially at air temperature.

True air temperature was measured by means of an aspirated thermocouple placed inside a thermos bottle (see Fig. 2*b*). The sampling hose was of short length so as to reduce the possibility of temperature change of the air between the sampling point and the thermocouple. A small air disturbance was created in the enclosure by the device but was found to be negligible in its effect on the thermocouples under test. Temperatures were measured with a probable accuracy of ±0.15 F. The combined sensitivity of the measuring equipment was about ±one microvolt (0.05 F).

The thermocouples were made from No. 36 AWG copper and No. 30 AWG constantan wires which had previously been carefully calibrated. The junctions were only slightly larger than the wire itself and were soft soldered.

All shields were two inches long and were constructed from aluminum sheets of 0.006 in. thickness. The shield construction is shown in Fig. 2*a*.

Stiff wire supports were passed through holes in the shields as shown in the figure and were glued at their intersections with each other and with the shields with a cement used in model airplane construction. The longitudinal joint of the shield was also cemented, and the whole structure was suspended vertically by gluing the thermocouple wire to the wire supports.

The source of these sheets is not known, but evidently they had had several years exposure to the atmosphere, as indicated by their oxidized appearance. However, they still retained a fair reflectivity for light and no attempt was made to polish them. Edwards [2] has shown that long exposure affects only slightly the reflectivity of aluminum foil as regards infra-red radiation. The

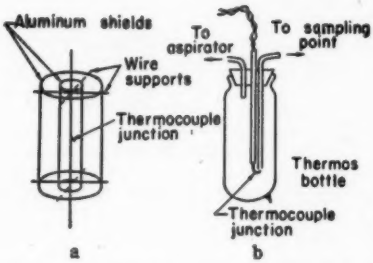

FIG. 2. *a* (*Left*) THERMOCOUPLE SHIELDS
b (*Right*) ASPIRATED THERMOCOUPLES

decrease in reflectivity for visible radiation is slightly greater. It is doubtful if appreciably different results would have been obtained by using more highly polished metal foil.

PROCEDURE

Each test consisted of a comparison between one shielded thermocouple and one unshielded couple with the result that no direct comparison between tests could be made on a temperature basis.

Temperature readings were taken first with the radiation source out of view of the opening in the enclosure and then continued with the source placed as shown in Fig. 1. After steady state conditions were reached, the hot body was removed and the temperature readings were continued for a short time thereafter. Temperature-time curves were plotted so as to determine the most representative temperature values for the test and also to establish the time response of the shielded and unshielded couples.

In order to compare the similarity of the various thermocouples used in the tests they were placed in the air blast from a small fan. Both shielded and unshielded couples then indicated exactly the same temperature.

[2] Some Reflection and Radiation Characteristics of Aluminum, by J. D. Edwards and C. S. Taylor, (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 179.)

OBSERVATIONS

The results obtained are given in Table 1. However, some discussion of the various factors involved can well be made here, together with certain observations which were made during the tests but which cannot readily be tabulated.

The temperature of an unshielded thermocouple is determined by the balance between radiation gain and convective heat loss if conduction from the junction

TABLE 1—SUMMARY OF TEST RESULTS

TEST NO.	1a	1b	2	3	4	5	6	7	8	9	10[a]
	Shielding Arrangements										
Number	3	1	1	1	1	1	2	2	2	2	3
Dia.—In.	½, ¾, 1	1	1	2	2	3	2, 3	1, 2	1, 2	1, 2	1, 2, 3
	Temperature Indicated by Thermocouples Deg F										
Aspirated	79.1	79.1	78.0	77.6	78.3	78.7	79.5	70.0	76.2	77.6	75.9
Shielded	81.6	80.6	79.5	78.6	79.6	79.9	80.3	71.8	77.6	78.9	77.1
Unshielded	81.2	81.2	79.8	79.0	80.7	80.4	80.8	72.6	78.4	79.9	77.8
	Error in Reading Referred to Aspirated Couple										
Unshielded											
Deg F	2.1	2.1	1.8	1.4	2.4	1.7	1.3	2.6	2.2	2.3	1.9
Per cent	2.7	2.7	2.3	1.8	3.1	2.2	1.6	3.7	2.9	3.0	2.5
Shielded											
Deg F	2.5	1.5	1.5	1.0	1.3	1.2	0.8	1.8	1.4	1.3	1.2
Per cent	3.2	1.9	1.9	1.3	1.7	1.5	1.0	2.6	1.8	1.7	1.6
Ratio of Error Shielded to Unshielded	1.2	0.71	0.83	0.72	0.54	0.71	0.62	0.69	0.64	0.57	0.62

[a] Average of five tests.

be neglected.[3] The radiant gain is affected by the temperature of the radiating objects and by the area and emissivity of thermocouple. The convective heat loss is affected by the convective area of the thermocouple, the temperature difference between it and the ambient air, the size of the junction, and the air velocity over it. By making the junction of very small size in order to increase the rate of convective heat loss, or by making it a good reflector of radiation, or by passing air over it at a high velocity, or by a combination of these methods, a thermocouple can be made to indicate very nearly the true air temperature. At best, however, a thermocouple can dissipate heat received by radiation only by means of a finite temperature difference, however small, between it and the air.

[3] Heat Transmission, by W. H. McAdams. (Second Edition, McGraw-Hill Publishing Co., New York, 1942, pp. 223-225.)

A single shield placed around the thermocouple reflects some of the incident radiation, the amount depending upon its surface character, and absorbs the rest. Part of this is reradiated to cooler objects including the thermocouple itself and the balance is dissipated by convection. If a series of concentric shields is used, the likelihood that heat will be radiated to the couple from the inner shield is reduced, while the convective area is increased, particularly if there is good conduction along the shields from the area receiving the radiation. Increasing the shield diameter tends to increase the ratio of convective area to radiation receiving area, but at the same time tends to decrease the unit rate of convective heat transfer. Increasing the length also reduces the unit rate of heat transfer. The actual construction required, of course, depends upon the nature of the radiating sources which may affect the thermocouple.

From theoretical considerations it would be expected that the temperatures of concentric shields would progressively decrease from outside to inside and that only in the ideal case would the innermost shield be at the same temperature as the thermocouple it enclosed. This was found to be true as shown by the following data taken in Test 6.

	TEMPERATURE DEG F
Outer shield	81.1
Inner shield	80.7
Shielded thermocouple	80.3
The air	79.5
Unshielded thermocouple	80.8

If concentric shields are placed too close together and/or the thermocouple junction is too close to the inner shield, shielding may become a positive detriment. This is borne out in part by the data of Test 1a where it will be seen that the shielded couple indicated a higher temperature than the unshielded couple. Evidently the junction was not only located in the convection current adjacent to the inner shield but also the temperature of this shield was greater than if the spacing of the three had been increased. This is in agreement with the well known fact that the conductance of an air space increases rapidly as the width of the space decreases. The small diameter shield in effect simply increases the diameter of the junction.

In the preliminary studies random air currents caused the galvanometer to fluctuate widely. With the instruments used these fluctuations were as great as ± 15 microvolts $\pm(0.7$ F). However, in a test made in direct sunlight indoors, the 1 in. O.D. triple shielded couple gave results similar to those listed under 1a of Table 1.

It was, therefore, found necessary to enclose the thermocouples under test and also to use the shields shown in Fig. 1 to prevent warm air currents generated by the radiant source from reaching the enclosure. There was later found to be an interchange of heat energy between the unshielded thermocouples and the outer shield of the shielded thermocouples. This effect was reduced by means of a cardboard barrier placed between them as shown by the dotted lines in Fig. 1. This shield was used in tests 2, 7, 8, and 9.

Depending upon the conditions within the test room, the aspirated couple gave temperature indications either equal to or below that of the junctions inside the cardboard enclosure. When the hot source was placed in front of

the enclosure, there was a lapse of about 15 min before there was a constant difference between the aspirated thermocouple and the others, which, of course, responded much more rapidly.

Readings of each were taken at definite time intervals and the data plotted on time-temperature coordinates. The data given in Table 1 were taken from these curves at the point where they became parallel. Therefore, in spite of the fact that the temperature level rose during a test due both to the radiant source and the building heating system, the results are considered to be reliable.

Included with the thermocouples of tests 1a and 1b (Table 1) was one made of calibrated No. 24 AWG wires. The junction was unshielded and, as would be expected, it indicated a temperature greater than that of the fine wire couple, in this case 0.4 F.

CONCLUSIONS

1. Random air currents can greatly affect the stability of readings of fine wire thermocouples and also those of wire as large as No. 24 gage. Use of a suitably sized shield seems desirable if for no other reason than to give stable readings.

2. In the absence of random air currents and in the presence of a body radiating at a temperature greater than that of the ambient air, the unshielded fine wire thermocouple indicated a temperature greater than the true air temperature.

3. The temperature of the shielded thermocouple which could see the radiant heat source was aways greater than the true air temperature.

(a) The concentric group of shields with the maximum diameter equal to one inch, gave thermocouple readings exceeding that of the unshielded fine wire thermocouple. This indicates that a temperature responsive element should be at least one-half inch away from the nearest shield.

(b) When a single shield of one inch diameter or greater was used, the per cent error decreased as the diameter of the shield increased. The error, referred to true air temperature, was for all such cases, less than that of an unshielded thermocouple. Based on the data of Table 1, the optimum diameter is about two inches.

(c) Two concentric shields with the inner diameter one inch or greater still further reduced the error of the thermocouple reading.

(d) Three concentric shields with the inner diameter one inch showed no improvement over either double concentric shield which was tested.

4. If it is necessary to determine the difference in temperature between a surface and the ambient air with a small percentage of error, even the use of shielding devices may not be satisfactory. If general air movement cannot be tolerated, an aspirated thermocouple seems to be the most satisfactory device to secure true air temperature.

DISCUSSION

C.-E. A. WINSLOW, New Haven, Conn.: We have been interested in this important matter in our laboratory and have found somewhat similar results in a room lined with aluminum foil where extreme radiation effects are produced. We found about a 2 deg excess due to radiation effects on the couples themselves. What the temperature of the surface is under these conditions is an interesting point. A thermocouple never gives the exact temperature of the surface, but something intermediate between the true temperature of the surface and the temperature of the air. In an extreme case, in which a room heated by reflected radiant heat from aluminum surfaces, with the walls and ceiling of aluminum, we found that the thermopile temperature, which is normally the most accurate indication, was something like 20 deg higher than the thermocouple temperature. The 20 deg higher figure represents the reflective effect of the aluminum surface but we

also found this other effect that is mentioned. If you want to obtain the true surface temperature, since it influences heat loss through the wall, a shielded thermocouple must be used.

C. M. ASHLEY, Syracuse, N. Y.: This paper calls attention to a most important subject of laboratory technique; one which I believe is very often overlooked. For example, in the laboratory of my company, we recently ran a test in which the indicated temperature was higher than actual by 10 deg due to radiant effects. Had that not been taken into account the interpretation of results would have been quite different from the actual interpretation. This paper should be recognized as being very definitely limited in scope. For instance, the interpretation of the shielding effect must be considered as good only under the conditions of the test because, obviously, under forced convection conditions, quite different shielding arrangements would give optimum results.

I would also mention that in my opinion under these conditions the aspirating type thermocouple is much to be preferred to a shielded thermocouple, since obviously the errors to be expected would be considerably less. Some possibilities of error are present in the measurement of the true air temperature by means of the thermometer in the thermos bottle.

DR. OSCAR IMALIS:[4] I wanted to ask how the true temperature was measured. The author spoke about the difference between shielded temperature, unshielded temperature of the thermocouple, and true air temperature.

F. W. HUTCHINSON, West Lafayette, Ind.: Mr. Parmelee mentioned that this work was done as part of a broader project involving research in glass, and Director Tasker almost apologized for the fact that they brought out this part by itself. I think he should be complimented upon doing so.

I want to call attention to the fact that it will be tremendously convenient in the future for many of you who are working with thermocouples to be able to find them under the word *thermocouples* in the TRANSACTIONS and not have to search, as we have had to do so frequently in the past, through glass research, panel heating research, and various other kinds of research in the hope of locating a small paragraph which tells about the instrument which has a much broader scope.

I think the policy of presenting brief papers on instrumentation is very valuable and highly effective.

M. K. FAHNESTOCK, Urbana, Ill.: The authors have pointed out that radiation shields applied to thermocouples used for measuring air temperatures can result in larger errors than if no shields are used. I just want to mention a simple method which we have used to obtain accurate air temperatures with unshielded thermocouples exposed to radiation levels often present in spaces or rooms occupied by human beings. This method makes use of corrections for radiation effects on unshielded thermocouples. The corrections are obtained by making observations consecutively in the respective positions with at least three different thermocouples made of three different sizes of wire. The larger the thermocouple, the larger will be the effect of the radiation. The reading obtained with each different size thermocouple plotted against its equivalent diameter will give a straight line which can be extended to zero equivalent diameter. The temperature corresponding to this zero equivalent diameter is the correct air temperature, and the difference between this value and that obtained with any one of the thermocouples is the correction for the radiation effect on that respective thermocouple.

This method of obtaining and using correction factors for radiation effects avoids the use of complicated shields and aspirated shields which are often impractical. The results will also probably be more accurate than those obtained with most shields.

[4] Former Research Physicist, A.S.H.V.E. Research Laboratory.

CYRIL TASKER, Cleveland, Ohio: I am very glad that Professor Hutchinson referred to the value of the short paper. However, this paper will be followed by similar short papers. Their success as a presentation will depend upon how much information we can get back from others working in the field as a result of presenting these short papers. We definitely invite your criticism, comment and help.

If I may say once again what I have been saying for two years now: this is your Laboratory and your program. You will help us if you will tell us of your experiences so that we can do your work a lot better.

MR. PARMELEE: I would like to answer Dr. Imalis' question first. The temperature of the room was not measured. What we measured was the temperature of the air within the enclosure, which is shown in Fig. 1. The temperature was measured by means of a thermocouple located, not in the space itself, but inside of a small thermos bottle (see Fig. *2b*). Connected to this thermos bottle was a short length of hose which drew air from the immediate vicinity of the shielded and unshielded thermocouples. As far as we were able to determine, this method obtained the air temperature within the enclosure.

Mr. Ashley has mentioned that under forced convection conditions the shielding construction need not necessarily be the same as that which we used. Obviously, there are a great many different methods of shielding that might be used. However, I think that if there is any appreciable air velocity, shielding is of less importance. Whether or not I can say that shielding is not required when we have substantial air motion might be questioned, but we did notice that when the unshielded thermocouple was located in the room, and sensitive apparatus was used, the indications on the galvanometer were not at all stable. We could not establish any one reading, and this would indicate that very small air currents may cause a thermocouple to register fairly close to true air temperature.

Professor Fahnestock has stated that one satisfactory method of measuring air temperature with thermocouples is to use three different sizes and by plotting a curve relating area to temperature indicated, determine the true temperature at zero area. In the particular study that we are conducting we anticipated having to make so many measurements that we thought it necessary to try to reduce to the minimum the number of observations that would be required. In the literature the shielded thermocouple is indicated as being a rather effective means of measuring temperature, but we have found that is not quite the complete solution of the problem.

1293

No. 1293

SEMI-ANNUAL MEETING, 1946

Montreal, Que., Canada

OVER 400 enthusiastic members and guests gathered in Montreal for the Semi-Annual meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June 10-13, 1946. Members began to arrive at the Mount Royal Hotel on Saturday and Sunday, June 8 and 9, in order to visit the historic and scenic attractions of Montreal. All arrangements for the Meeting were in charge of the Montreal Chapter members, who were on hand to greet the visitors.

The registration on Monday morning, June 10, at the Mount Royal Hotel was followed by a Welcome Luncheon in the Jacques Cartier Room at 12:15 p.m. F. A. Hamlet, Montreal Chapter, was toastmaster and introduced the national Society officers, Montreal chapter officers and members of the Committee on Arrangements.

Pres. A. J. Offner, New York, N. Y., responded to the welcome of A. B. Madden, Montreal Chapter president.

The principal speaker at the luncheon was His Worship, Camillien Houde, the Mayor of Montreal, whose hearty welcome and remarks were greatly appreciated.

FIRST SESSION—MONDAY, JUNE 10, 2:30 P.M.

The first technical session was called to order by President Offner in the Jacques Cartier Room at 2:30 p.m.

First Vice Pres. B. M. Woods, Berkeley, Calif., referred to the proposal made at the Annual Meeting in January, that the Council be authorized to invest such portion of the Society Endowment Fund as may be advisable for the purchase of real property for Society use and suggested that favorable action be taken.

President Offner stated that, since three of the four technical papers to be presented at the session were prepared from Society research or cooperative research, he would request Cyril Tasker, Cleveland, Ohio, director of research, to preside at the meeting and to introduce the authors and subjects.

G. V. Parmelee, Cleveland, Ohio, presented the paper Forced Convection Heat Transfer Coefficients Along a Flat Surface,[1] by Mr. Parmelee and

[1] Published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, May 1946.

R. G. Huebscher, which was a summary of a complete report of work carried out under the direction of the Technical Advisory Committee on Glass.

L. G. Seigel, Cleveland, Ohio, presented his paper, The Effect of Turbulence Promoters on Heat Transfer Coefficients for Water Flowing in Horizontal Tubes (see Chapter No. 1294). He stated that the paper was the result of work instigated by the Bureau of Ships, Navy Department, which desired to obtain a minimum weight and size for air conditioning equipment on shipboard. This investigation pertained to the use of turbulence promoters in $\frac{5}{8}$ in. O.D. tubes through which water was circulated as a cooling medium.

Dr. Brandt presented a summary of his paper Energy Losses at Suction Hoods, which he prepared with Russell J. Steffy as co-author (see Chapter No. 1295).

A. C. Stern,[2] New York N. Y., presented the paper Characteristics of Unit Dust Collectors, which he had prepared in conjunction with Jack Baliff, A. E. Perina, Robert Crowley, Benjamin Feiner, and A. A. Urbano (see Chapter No. 1296).

Mr. Stern stated that, in addition to describing methods of testing dust collectors, the paper showed that most of the units developed approximately 50 per cent of the manufacturers' ratings.

Second Vice Pres. G. L. Tuve then took charge of the meeting and announced that the *S.S. Quebec* would sail at 7:45 p.m. daylight saving time and that the second session would be held aboard the ship.

The meeting was adjourned at 5:20 p.m.

SECOND SESSION—TUESDAY, JUNE 11, 9:30 A.M.

The meeting was called to order in the music room of the *S.S. Quebec* by First Vice Pres. B. M. Woods, Berkeley, Calif., who introduced G. Lorne Wiggs, a member of Montreal Chapter. Mr. Wiggs presented his paper Air Conditioning a Canadian Windowless Textile Mill (see Chapter No. 1297).

C. O. Mackey, Ithaca, N. Y., then presented the paper, The Sol-Air Thermometer—A New Instrument, which he prepared with L. T. Wright, Jr., as co-author (see Chapter No. 1298).

Mr. Wright then presented the next paper Periodic Heat Flow—Composite Walls or Roofs, which he prepared with C. O. Mackey as co-author (see Chapter No. 1299).

The last paper of the session, Heat Loss Through Wetted Walls was presented by the author, E. C. Willey, Corvallis, Ore. (see Chapter No. 1300).

THIRD SESSION—WEDNESDAY, JUNE 12, 9:30 A.M.

The third technical session, which was held in the music room of the *S.S. Quebec* was called to order by Second Vice Pres. G. L. Tuve at 9:30 a.m., June 12.

The first paper, A Rational Basis for Solar Heating Analysis, by F. W. Hutchinson and W. P. Chapman, was presented by Professor Hutchinson, Lafayette, Ind. (see Chapter No. 1301). Professor Hutchinson pointed out

[2] Chief engineer, Division of Industrial Hygiene, New York State Department of Labor.

that the object of the study being made was to investigate solar heating under ideal conditions in actual houses built for test purposes in order to learn the advantages and disadvantages of solar heating.

Second Vice President Tuve announced that the New York and Pittsburgh Chapters had contributed $875 toward payment of the mortgage on the new Research Laboratory building.

First Vice President Woods announced that after further consideration by the Finance Committee and the Council it was found desirable to reconsider the action taken at the first technical session regarding the Endowment Fund and after a motion to reconsider the action was voted, a resolution to withdraw the original motion was adopted.

C. C. Wright, State College, Pa., presented the paper Hold-Fire Controls for Bituminous Coal Stokers, which he prepared with T. S. Spicer and H. A. Baumann as co-authors (see Chapter No. 1302).

Dr. C.-E. A. Winslow, New Haven, Conn., presented the next paper on The Influence of an Interior Coating of Aluminum Paper on Internal Thermal Conditions and Heat Economy, which he prepared with R. J. Lorenzi and L. P. Herrington (see Chapter No. 1303).

ENTERTAINMENT

The first combined land and sea meeting of the Society provided an opportunity to visit in a neighboring country with pleasant relaxation aboard the *S.S. Quebec.*

While Committee and Council members attended business sessions on Sunday and Monday morning June 9-10, others were able to do some sightseeing. This included a drive to the top of Mount Royal for a view of the city and of the St. Lawrence River, and visits to churches, cathedrals and shrines. In addition, shopping trips were enjoyed by the ladies on Monday, both before and after the Welcome Luncheon.

Immediately after the Monday technical session members and guests boarded the *S.S. Quebec.* Music for dancing and entertainment was provided in the music room until 12:00 o'clock each evening.

On Tuesday afternoon, a stop at Murray Bay provided an opportunity for a visit to the Manoir Richelieu, from which many members brought back purchases of interesting Canadian handiwork. A drenching rain prevented the sports events which had been scheduled by the committee.

Some of the ladies took this occasion to inspect the ship's galley while members of the Society accepted an invitation to inspect the engine rooms.

On Tuesday evening after a brief stop at Tadoussac, the *S.S. Quebec* entered the fjords of the Saguenay, the deepest river in the world. The ship arrived at Bagotville, the head of navigation for ocean going ships, during the night.

On Wednesday morning many members were able to watch the sunrise over the Laurentian Mountains. Later, as the ship passed the capes of the Saguenay on its return voyage the ship's engines were stopped to permit taking of pictures and for enjoyment of the beauty of the thousand foot cliffs.

A return stop at Tadoussac permitted members to visit the oldest fur trading post in North America.

BANQUET

The *S.S. Quebec* docked at Quebec at 6:30 p.m. and members and guests were taken to the Chateau Frontenac for the Semi-Annual banquet. F. A. Hamlet, Montreal, was toastmaster at the banquet.

Speeches were made by Monsieur M. Viau, representing His Worship the Mayor of Quebec, Lucien Borne, and the Honorable Paul Balyou, Minister of Trade and Commerce, who represented the Prime Minister of Quebec Province.

Responses to the fine sentiments expressed by the Canadian speakers were given by President Offner and President Madden of Montreal Chapter.

Eight past presidents of the Society were present: M. F. Blankin, Philadelphia, Pa.; W. H. Carrier, Syracuse, N. Y.; S. H. Downs, Kalamazoo, Mich.; H. P. Gant, Glenmore, Pa.; E. Holt Gurney, Toronto, Ont.; C. V. Haynes, Portland, Me.; W. T. Jones, Boston, Mass., and C.-E. A. Winslow, New Haven, Conn.

Past President Gurney made the presentation speech on the memory book given to Dr. Winslow, as president of the Society in 1945. The Committee on Resolutions presented the following resolutions, which were adopted unanimously:

RESOLUTIONS

The Resolutions Committee appointed by President Offner offers the following:

WHEREAS, the great success of the 1946 Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been due to the careful and thorough preparations of the Montreal Chapter and affiliate Canadian Chapters, and

WHEREAS, the active effort of the Committee on Arrangements, under the able leadership of its General Chairman, F. A. Hamlet, and Leo Garneau, Vice-Chairman, supported by A. B. Madden, President of the Montreal Chapter, together with His Worship, the Mayor of Montreal,

●

BE IT RESOLVED, that an expression of appreciation be adopted and that copies be sent to them and to each of the following:

To the Management and Staff of the Mount Royal Hotel for their able assistance;

To the Management and Staff of the Chateau Frontenac for their gracious hospitality;

To the Montreal Convention and Tourist Bureau for the tours and entertainment at Montreal;

To the Newspapers and Trade press of both Canada and the United States;

To M. Viau who represented His Worship the Mayor of Quebec, Lucien Borne, and to Honorable Paul Balyou, Minister of Trade and Commerce in the Provincial Government of Quebec representing the Prime Minister of the Province, for their participation in making the Banquet the climax of this very successful Society Summer Meeting;

●

To the Canada Steamship Co., especially Messrs. Bonter, Daly, Fisher and Captain Burch and all of his Staff who have, under these difficult times, rendered us such outstanding service and cooperation;

To the Speakers for the time and effort necessary to prepare the technical papers which were very well received and enjoyed;

To the eight past presidents of the Society who are with us at this Meeting. These resolutions are offered for adoption.

Respectfully submitted,
Carl F. Boester, *Chairman*
L. T. Avery W. E. Heibel
J. E. Haines F. E. P. Klages
G. W. F. Myers

After the banquet, members and guests returned to the *S.S. Quebec* which sailed for Montreal at 11:00 p.m.

Although wind and tide delayed arrival in Montreal Thursday morning, the 13th, it was the consensus of all who made the cruise that they had a wonderful trip and had enjoyed one of the best summer meetings ever held by the Society.

PROGRAM SEMI-ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

JUNE 10-13, 1946

Sunday, June 9

9:00 A.M. Reception of Members and Ladies
10:30 A.M. Guide Publication Committee

Monday, June 10

9:00 A.M. REGISTRATION—(Mount Royal Hotel, Montreal)
10:00 A.M. Council Meeting
12:15 P.M. Welcome Luncheon (Mount Royal Hotel—Jacques Cartier Room) for Members, Ladies and Guests
SPEAKER: Camillien Houde, His Worship the Mayor of Montreal
2:00 P.M. FIRST SESSION—(Mount Royal Hotel—Jacques Cartier Room)
Pres. Alfred J. Offner presiding
Greetings by Pres. A. B. Madden, Montreal Chapter
Response by President Offner
Committee Reports
Forced Convection Heat Transfer Coefficients Along a Flat Surface by G. V. Parmelee and R. G. Huebscher
The Effect of Turbulence Promoters on Heat Transfer Coefficients for Water Flowing in Horizontal Tubes by L. G. Seigel
Energy Losses at Suction Hoods by A. D. Brandt and R. J. Steffy
Characteristics of Unit Dust Collectors by A. C. Stern, Jack Baliff, A. E. Perina, Robert Crowley, Benjamin Feiner and A. A. Urbano
2:15 P.M. Ladies' Sightseeing Trip and Shopping Tour (two hours)
5:30 P.M. All Aboard *S.S. Quebec* for Cruise on St. Lawrence River
6:45 P.M. Left Montreal
7:00 P.M. First Call for Dinner
8:30 P.M. Entertainment and Dancing in Music Room of Steamer

Tuesday, June 11

7:30 A.M. *S.S. Quebec* Left Quebec for Murray Bay

9:30 A.M. SECOND SESSION—(*S.S. Quebec*—Music Room)

Air Conditioning a Canadian Windowless Textile Mill by G. Lorne Wiggs

The Sol-Air Thermometer—A New Instrument, by C. O. Mackey and L. T. Wright, Jr.

Periodic Heat Flow—Composite Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr.

Heat Losses Through Wetted Walls, by E. C. Willey

12:30 P.M. Arrived Murray Bay for Sightseeing, Swimming, Golf, Tennis at Manoir Richelieu

3:30 P.M. All Aboard for the Saguenay

4:00 P.M. Chapter Delegates Meeting

7:00 P.M. Sightseeing at Tadoussac

9:00 P.M. Entertainment and Dancing, Travelog Films of the Province of Quebec

Wednesday, June 12

1:00 A.M. Left Bagotville

9:30 A.M. THIRD SESSION—(*S.S. Quebec*—Music Room)

A Rational Basis for Solar Heating Analysis by F. W. Hutchinson and W. P. Chapman

Hold-Fire Controls for Bituminous Coal Stokers by C. C. Wright, T. S. Spicer and H. A. Baumann

The Influence of an Interior Coating of Aluminum Paper in Internal Thermal Conditions and Heat Economy by R. J. Lorenzi, L. P. Herrington and C.-E. A. Winslow

2:00 P.M. Deck Sports and Games

2:30 P.M. Nominating Committee Meeting

7:00 P.M. Arrived Quebec

7:30 P.M. Banquet (Chateau Frontenac, Quebec)

SPEAKER: His Worship the Mayor of Quebec

French Canadian Folklore Music

10:30 P.M. Left Quebec

Thursday, June 13

8:30 A.M. Arrived Montreal

COMMITTEE ON ARRANGEMENTS

F. A. HAMLET, *General Chairman*

LEO GARNEAU, *Vice-Chairman*

Banquet—A. B. MADDEN, *Chairman;* J. B. FLANAGAN, G. L. BALLANTYNE

Entertainment—T. H. WORTHINGTON, *Chairman;* A. F. LAMONTAGNE, JACK WRIGHT

Finance—F. G. PHIPPS, *Chairman;* J. D. ROSS, C. A. BOOTH

Inspection—A. M. PEART, *Chairman;* J. G. CHENEVERT, W. J. ORR

Ladies—W. W. TIMMINS, *Chairman;* A. E. WATTS, F. A. SHEPPARD

Publicity—W. G. HOLE, *Chairman;* E. H. TERRANCE, J. H. STOREY

Sessions—J. B. DYKES, *Chairman;* B. J. HORSBURGH, RALPH GROSSMAN

Reception—J. J. COSGROVE, *Chairman;* J. P. FITZSIMONS, JOHN COLFORD

Secretary—S. W. SALTER

1294

No. 1294

THE EFFECT OF TURBULENCE PROMOTERS ON HEAT TRANSFER COEFFICIENTS FOR WATER FLOWING IN HORIZONTAL TUBES

By LAWRENCE G. SEIGEL,* CLEVELAND, OHIO

This paper is a result of research work at Case School of Applied Science at the instigation of the Air Conditioning Section, Bureau of Ships, Navy Department.

INTRODUCTION

TURBULENCE promoters or retarders have been used for some time to increase the rate of heat transfer from a given amount of surface. However, little information has been available on the relative merits of the various types of devices which have been used. Colburn and King [1] have given data on turbulence promoters for air flowing inside 2⅝ in. steel tubes and somewhat similar data, though not so complete, have been given by Royds [2] for 2$^{29}/_{32}$ in. tubes. Nagaoka and Watanabe [3] have investigated the use of turbulence promoters with water flowing in 1 in. pipes, but no information was found for liquids in small diameter tubes. Data are lacking also on film coefficients for water flowing in the laminar region at low Reynolds numbers.

This investigation was undertaken to establish accurate information in regard to the increase in heat transfer and pressure drop which would result from the use of some of the common turbulator designs in ⅝ in. outside diameter copper tubes. *All the data covered by this paper are for heating water at low temperatures in horizontal copper tubes of ⅝ in. outside diameter.* Therefore, they may be applied directly to heat exchangers similar to chilled water cooling coils.

TEST METHODS AND PROCEDURE

Since the results of this study were to be applied directly to air cooling coils using chilled water as a refrigerant, it was thought best to study the effect of the turbulence promoters in samples of coils operating under selected

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* Professor in Mechanical Engineering, Case School of Applied Science. Member of A.S.H.V.E.

[1] Exponent numerals refer to Bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Montreal, Que., Canada, June, 1946.

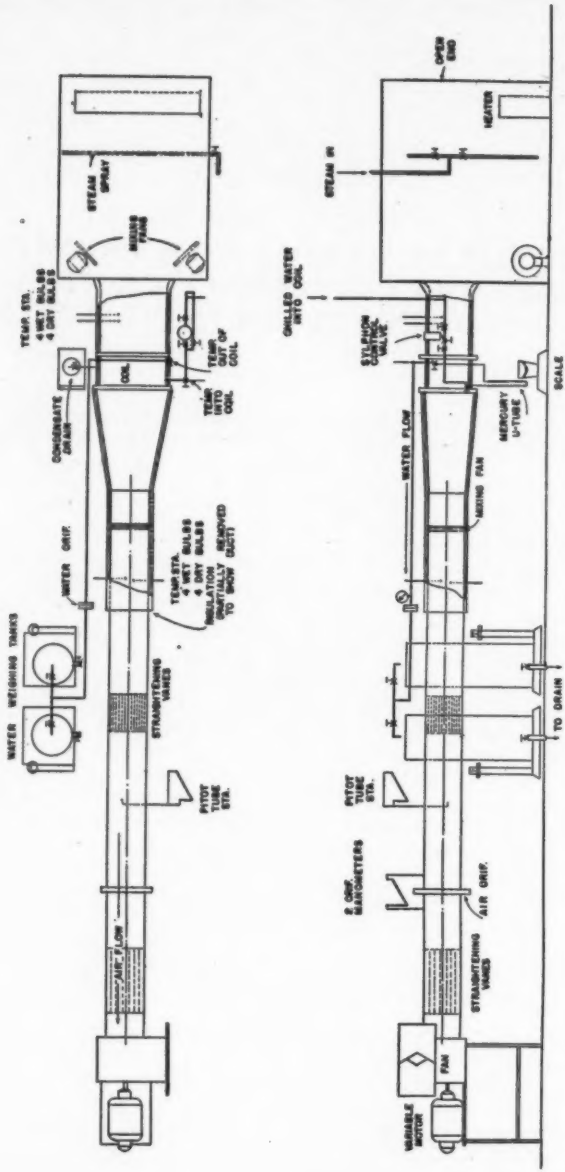

FIG. 1. SETUP FOR TESTING WATER COILS

load conditions. Accordingly, a full size air conditioning coil was obtained for the purpose. This coil was of the plate fin type with 8 rows of 5/8 in. outside diameter copper tubes. Water circuiting was arranged as nearly counterflow with respect to the air as possible. Return bends on one end of the coil were soft-soldered so that turbulators could be changed when necessary. The coil was installed in a duct equipped with a preconditioning chamber upstream and a mixing device and air flow orifice downstream as shown in Fig. 1. Air was drawn through the system by a large variable speed blower. Air temperatures were obtained from precision thermometers located at equally spaced points around the duct both upstream and downstream. Four wet bulb thermometers and four dry bulb thermometers were located at each station and readings were taken at intervals of five minutes for at least one hour after equilibrium had been established. The air orifice was calibrated in place by means of a pitot tube. Water quantities were obtained by direct weight and water pressure

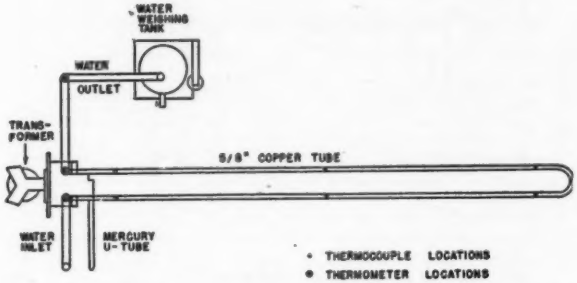

FIG. 2. DIAGRAM OF ELECTRICALLY HEATED TUBE FOR DETERMINING FILM COEFFICIENTS

drop was measured by a mercury manometer connected across the coil headers. Heat balance checks between the air and water showed agreement within plus or minus 5 per cent for all tests and for most tests showed agreement within plus or minus 2 per cent.

In order to isolate the effect of the turbulence promoters in these tests, film coefficients of heat transfer were evaluated for the water side of the coil. The method used in calculating these coefficients was based on principles set forth in another paper,[4] and the calculations were extremely tedious because of the fact that it was necessary to analyze the coil performance on a row-by-row basis. No complete computation for any test will be given here because of the extreme length of such an example. Tests of the free tubes (i.e., tubes without turbulators) and of three types of turbulators were run and calculated by these methods, but because of the procedure and calculations involved it was decided that some simpler method of obtaining comparative data on film coefficients should be found.

For this purpose two 10 ft lengths of 5/8 in. outside diameter refrigeration tubing, connected by a return bend, were arranged in a low voltage electrical

circuit as shown in Fig. 2. The current used in the tube was varied from 50 to 800 amperes depending on the water quantity flowing in the tube. Tube temperatures were measured by thermocouples attached to the tube and water temperatures were determined by matched precision thermometers. The thermocouples were soldered to the outside surface of the tube after preliminary checks proved that this method of attachment gave consistent results with other methods such as inserting the couples in saw cuts or peening them in the tube surface. The tube was placed in a box and covered with expanded mica for thermal insulation. The electrical current input was measured but no

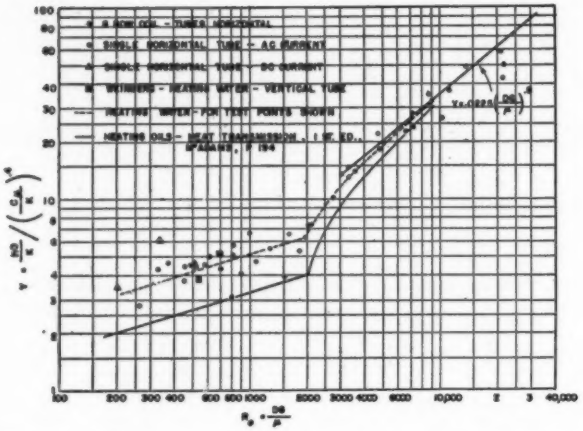

FIG. 3. CORRELATION OF HEAT TRANSFER DATA FOR WATER FLOWING INSIDE PIPES

attempt was made to use this data to calculate heat quantities because of the high electrical losses occurring in the connections. For determining coefficients from this setup, heat transfer was based on the water quantity times its temperature difference and the temperature drop from tube to water was taken as the arithmetic mean between the tube and water. The temperature drop through the tube wall was neglected since this amounted to less than 0.02 Fahrenheit degrees in most cases.

HEAT TRANSFER RESULTS

Fig. 3 shows a correlation of all heat transfer test data for free tubes. This includes data obtained by use of both experimental setups described. Agreement of points obtained by use of these two setups is significant since the calculation procedures used to evaluate the coefficients were different for each test method. In the full scale coil tests no surface temperatures were measured and the tube to water temperature differences were determined on the basis

of a computed surface temperature. Agreement of these coefficients with the results of the tests in which surface (or tube) temperatures were measured indicates that the *computed surface temperatures are reliable.* This is important because any coil performance calculation procedure using tube film coefficients as basic data must also be capable of evaluating satisfactory surface temperatures for the application of these coefficients.

Fig. 3 also shows agreement with other investigators as indicated by the dotted line in the turbulent region. In the laminar region few data for water in horizontal tubes are available for comparison. In general the correlation of this paper for laminar flow is about 60 per cent above the accepted line for oils which is shown by the lower solid line in Fig. 3. Fair agreement was

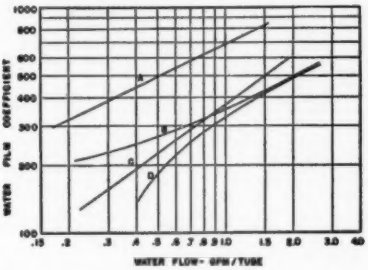

FIG. 4. COMPARISON OF WATER FILM COEFFICIENTS FOR 5/8 IN. O.D. COPPER TUBE WITH VARIOUS TYPES OF PROMOTERS

A—Steel spiral, 3/4 in. pitch, 12 gage wire; B—Twisted copper strip, 3 in. pitch, 0.02 in. thick; C—3/4 in. copper tube, sealed ends; D—Free tube

obtained with the data of Weinberg [5] for water flowing upward in vertical tubes. However, only two points calculated from Weinberg's data had sufficiently low Reynolds numbers to show on the curve.

Since the coefficients in the laminar region seemed high, two questions arose as to the accuracy of the test procedure. First it was thought that the alternating current used for heating the tube might have set up vibrations which increased the turbulence. This was checked by substitution of direct current for some tests but no significant change of coefficient was found. Results calculated from the data of these check tests are indicated by triangles in Fig. 3. Next it was noted that the tube temperature at the bend was inconsistent during laminar flow and it was assumed that the bend might have caused an increase in turbulence in the second ten foot tube. This was investigated by removing the bend and returning the water to the second tube by exactly the same piping arrangement by which it entered the first. Coefficients were then calculated for each tube separately and agreement with the points of Fig. 3 was still obtained.

It would be desirable to determine coefficients for oils with this setup to see if the cause for disagreement lies in the method or in the evaluation of the properties of the fluid. One other possible cause for the disagreement in the laminar region may be the fact that no calming sections were used before the tubes in these tests although they have been commonly used by other investigators.

A comparison of the coefficients obtained for the free tubes with those obtained by use of various types of turbulators is shown in Fig. 4. (The turbulators used in tests are shown in Fig. 5.) These results have been plotted against gallons per minute per tube instead of Reynolds number because of the

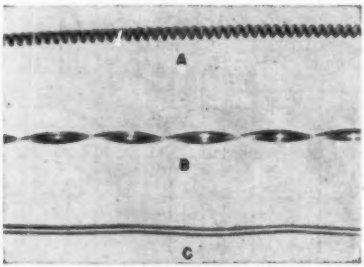

FIG. 5. GENERAL TYPES OF TURBULATORS
A—Spiral spring type, various pitches and wire diameters; B—Twisted copper strip 0.02 in. thick, 3 in. pitch; C—$\frac{3}{4}$ in. copper tube, sealed ends

difficulty of evaluating the Reynolds number when the turbulators were used. Test points have been omitted from these curves to avoid confusion. However, the points are shown in Fig. 6 for the spiral type turbulators in comparison with the free tubes.

Since the spiral springs similar to type A in Fig. 5 showed the best results, several sets of this type were tried. This was done to determine the effect of the distance between turns on the heat transfer coefficient and the pressure drop. No change in heat transfer coefficient could be found when the distance between turns was varied from $\frac{9}{16}$ in. to $\frac{3}{4}$ in. However, the pressure drop decreased with increased spacing. In an effort to obtain the optimum spacing for low pressure drop with maximum heat transfer, tests are being continued to determine what maximum spacing between turns may be used.

PRESSURE DROP

While it may be possible to correlate pressure drop and heat transfer to determine the *best turbulator for a given heat exchanger,* no general correlation can be made which applies to all exchangers. In some cases the pressure drop of the connecting piping may be extremely large compared with that of the heat transfer equipment and, therefore, an increase of several hundred per cent in equipment pressure drop may not be excessive. However, other

cases may permit only slight increase in pressure drop of the equipment before pumping costs rise beyond savings of space and equipment. Complete details of this pressure drop will be given in a subsequent paper.

CONCLUSIONS

From the results of this investigation the following items may be concluded:

1. The data presented are in fair agreement with that of other investigators, but more work by others for water in laminar flow would be desirable to check the results presented here. While practical water quantities are usually well within the

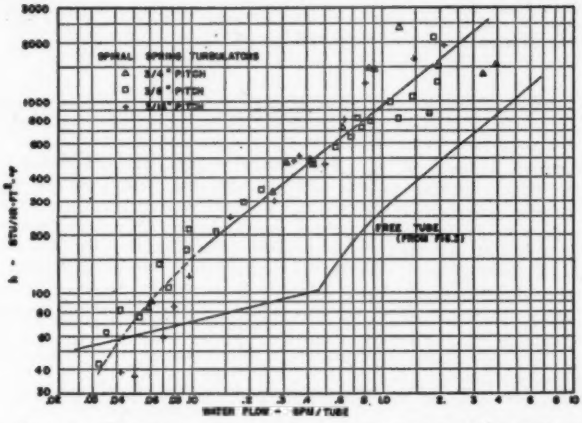

FIG. 6. EFFECT OF SPIRAL SPRING TURBULATORS ON HEAT TRANSFER COEFFICIENTS

turbulent region, occasions may arise when laminar flow exists and adequate data therefor should be available.

2. The method of calculating coil performance by evaluating a surface temperature as suggested in Bibliography [4] is satisfactory since heat transfer coefficients determined by use of this method are consistent with those obtained by other calculation methods and by other investigators.

3. The use of turbulence promoters increases heat transfer at the expense of pressure drop. This increased pressure drop is localized within the heat exchanger and therefore may not be serious when considered in terms of the pressure drop of the entire system.

4. Spiral spring turbulators like type A of Fig. 5 are of simple construction and their use results in a large increase in heat transfer rate. More work is required, however, to establish the proper ratio of coil spacing to wire diameter.

5. The use of turbulence promoters should be studied in direct expansion systems as well as in systems using other fluids.

ACKNOWLEDGMENTS

Thanks is extended to Prof. G. L. Tuve for his constructive criticism and helpful suggestions and to the following members of the research staff of Case

School of Applied Science who performed the tests and carried out the details of calculations: R. K. Ehrenfeld, S. R. Rubin, and R. D. Mandt.

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2. Heat Transmission by Radiation, Conduction and Convection by R. Royds, p. 190. (Constable & Co. Ltd., London, 1921.)
3. *Proceedings*, International Congress on Refrigeration, (7th Cong.), vol. 3, no. 16, pp. 221-245, 1937, J. Nagaoka and A. Watanabe. (The Hague, Amsterdam, 1936.)
4. A.S.H.V.E. Research Report No. 1271—Air Cooling Coil Problems and Their Solution, by L. G. Seigel. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 165.)
5. *Transactions*, American Institute of Chemical Engineers, 1942, vol. 38, pp. 493-530, R. C. Martinelli, C. J. Southworth, et al.

DISCUSSION

C. M. ASHLEY, Syracuse, N. Y.: The work done by Professor Seigel has added materially to the knowledge in this slightly explored field. In the past, many makers of tubular type heat transfer surface have considered the use of turbulence promoters. However, generally it has been found that the desired increase in transfer could be effected with less increase in pressure drop by changing the flow circuit in the heat transfer surface so that the water velocity through bare tubes was increased. Another objection is the difficulty involved in the use of turbulence promoters where fouling may occur. For these reasons, turbulence promoters are hardly ever used today except for very special applications.

I wonder if Professor Seigel can give us an interpretation of the results of his tests in terms of heat transfer *vs.* pressure drop in comparison with a plain tube. It would also be of interest to know whether Professor Seigel has any explanation for the scattering of data as plotted.

W. R. HEATH, Buffalo, N. Y.: Was any attempt made to attach the turbulator to the tubes to increase the total transmission by increasing the internal surface as well as by increasing the turbulence?

If not, is such an arrangement contemplated for future tests?

H. J. RYAN, New York, N. Y.: Might not an increase in pressure drop due to the use of turbulence promoters be out of proportion to the gain in heat transfer obtained.

G. L. TUVE, Cleveland, Ohio: I recall one instance in which a 30 per cent increase of cooling effect was obtained in a test room by bending of the cooling coil tubing into a spiral shape. I might also add that we invite any correspondence relating to the use of turbulence promoters because much research work is now under way on the use of turbulators in refrigerant evaporator coils.

PROFESSOR SEIGEL: The cause of scattering of points has not been found, but a similar condition has been noted in the results of tests using oil as a cooling medium, as reported in the first edition of *Heat Transmission* by William H. McAdams.

Referring to Mr. Heath's question the turbulence promoters fitted snugly in the tubes but were not attached thereto.

1295

No. 1295

ENERGY LOSSES AT SUCTION HOODS

By ALLEN D. BRANDT * AND RUSSELL J. STEFFY **

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, at its Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio, in cooperation with the U. S. Public Health Service.

PURPOSE OF STUDY

THE energy loss at suction hoods is an important consideration in the design of local exhaust systems for industrial atmospheric sanitation.[1-4] Since this loss can be minimized by proper hood design, it has been the tendency among designers in recent years to shape the hoods in such manner that this loss would be as small as practically possible. On the basis of existing data, this procedure necessitated long transformation sections from the hoods or hood faces to the ducts and frequently made the design needlessly difficult owing to interference with parts of the equipment or machine.

Recent field observations indicated that the energy loss at hoods is probably not so high as indicated in the literature. If these observations are correct, it would not be necessary to use long transformation sections and, as a result, hood design would be less difficult and less material would be needed for hood construction.

In addition, the static suction in a duct close to an exhaust hood frequently affords the most convenient, if not the only practicable method of estimating the quantity of air being exhausted from a given operation.[5] This information is essential to industrial hygiene engineers since the degree of contamination control is a function of the quantity of air exhausted. It is also extremely important that the relationship between the exhaust ventilation rate and the contaminant concentration in the atmosphere be known for different hoods and different operations if it is to serve as a guide on other similar installations. If, as indicated by field observations, most energy losses at hoods reported in the literature [1-8] are too high, an avoidable error is introduced into the ventilation rate.

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* Sanitary Engineer, U. S. Public Health Service, assigned to A.S.H.V.E. Research Laboratory. Member A.S.H.V.E.

** Research Assistant, A.S.H.V.E. Research Laboratory.

[1] Exponent numerals refer to Bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Montreal, Que., Canada, June 1946.

Consequently, it was deemed advisable to study the energy loss at a large variety of hoods to determine as accurately as practicable the true values and to provide information which would serve to estimate with a fair degree of accuracy the coefficient of entry for any hood.

EXPERIMENTAL PROCEDURE

The experimental procedure and a discussion of the types of hoods investigated in this study are covered in detail in the Appendix and will be summarized only very briefly at this point.

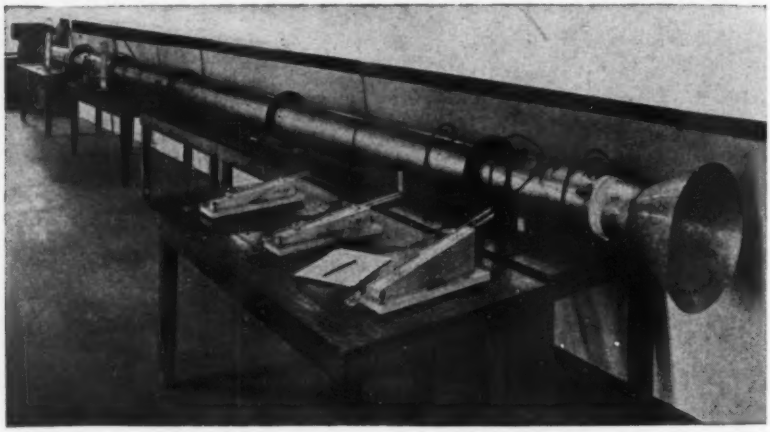

FIG. 1. SETUP FOR TESTING SUCTION HOODS

A total of 175 different hoods, including (1) different shapes, (2) different tapers or included angles, (3) different throat sizes, (4) different ratios of face area to throat area, (5) flanged and unflanged types and (6) different flange sizes, were tested at four different rates of air flow. The experimental setup is shown in Fig. 1, which is self-explanatory (see Appendix for more detail).

FINDINGS

The coefficients of entry for all hoods were computed by the formula

$$C = \frac{Q}{4008A\sqrt{h_s}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where C = coefficient of entry
Q = measured quantity of air being exhausted
A = area of the duct in which h_s was determined
h_s = measured static suction in inches of water

This is the same as the more common expression which follows since

$$Q = 4008A\sqrt{h_v}$$

$$f = \sqrt{\frac{h_v}{h_s}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

where f = coefficient of entry
h_v = duct velocity pressure in inches of water
h_s = static suction in duct near hood in inches of water

All authors do not agree on what is meant by the *coefficient of entry*. To industrial hygiene engineers the coefficient of entry expresses the ratio of the actual air flow into a hood to the theoretical air flow as indicated in the foregoing equations. One author,[9] on the other hand, defines the coefficient of entry as the ratio of the average velocity over the full area to the actual velocity in the vena contracta. Since there is considerable static regain downstream from the vena contracta, the coefficients reported by this author are very low. For the purposes of design of local exhaust systems and measurement of air quantities exhausted from given operations, the ratio of the actual to the theoretical air flow rate has much more value than other expressions. The terms *energy loss* or *entrance loss* and *coefficient of entry* are used more or less interchangeably throughout this paper. This may be confusing to some readers unless it is remembered that the energy loss (resistance or pressure drop) is a function of the coefficient of entry squared as shown in Equation (3).

$$h_e = \left(\frac{1 - C^2}{C^2}\right) h_v \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

where h_e = energy loss in inches of water
C = coefficient of entry
h_v = duct velocity pressure in inches of water

In Table 1 are summarized the coefficients of entry for all of the 167 8-in. diameter hoods studied. The hoods listed include round, rectangular, square, flanged and unflanged types. In addition to these hoods the investigation included a few having different diameter throats but which were otherwise proportional. These will be discussed separately later. The hood identification used in this and other tables is significant as explained more fully in the Appendix. The number (or numbers) before the first dash identifies the hood group, the one following the dash the approximate hood face area to throat area ratio. F indicates a flanged hood with 12 in. wide flanges unless otherwise specified, the number following the second dash indicates the flange width in inches if other than 12 in., and the R indicates a circular flange (most flanges on circular hoods were square).

A study of Table 1 will show that only 17 of the hoods tested have coefficients of entry below 0.850. Of these, 14 (groups 18, 19 and 16-F-½R) represent some variety of an open duct end and 2 (1-1.2 and 16-1.2) have unusually low face area to throat area ratios. The majority of the coefficients (62 per cent) are 0.90 or above and of a total of 82 unflanged hoods studied 26, or 32 per cent, had coefficients of 0.950 or higher. It is apparent that, with but few exceptions, the results obtained in this study are higher in general than those commonly reported in the literature. Notable among the exceptions

TABLE 1—SUMMARY OF TEST RESULTS FOR HOODS HAVING 8 IN. THROAT DIAMETERS
(*Duct velocity 4000 fpm*)

Hood Number	Shape	Included Angle* Deg	Coefficient of Entry	Hood Number	Shape	Included Angle* Deg	Coefficient of Entry	Hood Number	Shape	Included Angle* Deg	Coefficient of Entry
1-2.5	Round	10	0.926	8-4	Square	40	0.944	14-5F-1	Rect.	107	0.883
1-2.5F	Round	10	0.967	8-3	Square	40	0.942	15-5	Round	136	0.883
1-2	Round	10	0.922	8-3F	Square	40	0.950	15-5F	Round	136	0.895
1-1.6	Round	10	0.888	8-3F-8	Square	40	0.955	15-4	Round	136	0.893
1-1.2	Round	10	0.823	8-3F-4	Square	40	0.952	15-3	Round	136	0.895
2-2.5	Rect.	17	0.933	8-3F-2	Square	40	0.950	15-3F	Round	136	0.891
2-2.5F	Rect.	17	0.968	8-2	Square	40	0.933	15-2	Round	136	0.902
2-2	Rect.	17	0.925	8-2F	Square	40	0.938	15-2F	Round	136	0.885
2-1.6	Rect.	17	0.903	8-2F-8	Square	40	0.939	16-5	Round	90	0.948
3-6	Square	90	0.901	8-2F-4	Square	40	0.941	16-5F	Round	90	0.955
3-6F	Square	90	0.908	8-2F-2	Square	40	0.941	16-4	Round	90	0.952
3-5	Square	90	0.895	8-2F-1	Square	40	0.942	16-3	Round	90	0.950
3-4	Square	90	0.894	9-5	Rect.	29	0.969	16-3F	Round	90	0.955
3-3	Square	90	0.898	9-5F	Rect.	29	0.981	16-2	Round	90	0.952
3-3F	Square	90	0.900	9-4	Rect.	29	0.962	16-2F	Round	90	0.947
3-2	Square	90	0.887	9-3	Rect.	29	0.952	16-1.6	Round	90	0.962
3-2F	Square	90	0.902	9-3F	Rect.	29	0.966	16-1.4	Round	90	0.957
4-5	Rect.	109	0.870	9-2	Rect.	29	0.938	16-1.2	Round	90	0.811
4-5F	Rect.	109	0.872	9-2F	Rect.	29	0.960	16-F-½R	Round	90	0.839
4-4	Rect.	109	0.867	10-5	Rect.	81	0.936	17-5	Round	21	0.967
4-3	Rect.	109	0.858	10-5F	Rect.	81	0.952	17-5F	Round	21	0.983
4-3F	Rect.	109	0.873	10-4	Rect.	81	0.939	17-4	Round	21	0.963
4-3F-4	Rect.	109	0.876	10-3	Rect.	81	0.926	17-3	Round	21	0.960
4-3F-2	Rect.	109	0.880	10-3F	Rect.	81	0.943	17-3F	Round	21	0.975
4-2	Rect.	109	0.845	10-2	Rect.	81	0.912	17-2	Round	21	0.946
4-2F	Rect.	109	0.879	10-2F	Rect.	81	0.950	17-2F	Round	21	0.964
5-5	Rect.	134	0.860	11-5	Round	58	0.971	18-1	Round	0	0.717
5-5F	Rect.	134	0.883	11-5F	Round	58	0.978	18-F-10	Round	180	0.803
5-4	Rect.	134	0.862	11-5F-8	Round	58	0.979	18-F-8	Round	180	0.811
5-3	Rect.	134	0.855	11-5F-4	Round	58	0.974	18-F-4	Round	180	0.814
5-3F	Rect.	134	0.886	11-5F-2	Round	58	0.973	18-F-2	Round	180	0.818
5-3F-8	Rect.	134	0.888	11-5F-1	Round	58	0.974	18-F-1	Round	180	0.835
5-3F-4	Rect.	134	0.884	11-5F-1R	Round	58	0.978	18-F-1R	Round	180	0.832
5-3F-2	Rect.	134	0.890	12-6	Square	58	0.910	19-1	Square	0	0.697
5-2	Rect.	134	0.858	12-6F	Square	58	0.923	19-F	Square	180	0.791
5-2F	Rect.	134	0.892	12-6F-8	Square	58	0.922	19-F-8	Square	180	0.795
5-2F-8	Rect.	134	0.895	12-6F-4	Square	58	0.923	19-F-4	Square	180	0.792
5-2F-4	Rect.	134	0.892	12-6F-2	Square	58	0.924	19-F-2	Square	180	0.795
5-2F-2	Rect.	134	0.894	12-5	Square	58	0.919	19-F-1	Square	180	0.800
5-2F-1	Rect.	134	0.888	12-4	Square	58	0.922	20-5	Round	50	0.980
6-5	Rect.	54	0.950	12-3	Square	58	0.916	20-4	Round	50	0.978
6-5F	Rect.	54	0.960	12-2	Square	58	0.895	20-3	Round	50	0.973
6-5F-8	Rect.	54	0.949	12-2F-2	Square	58	0.905	20-2	Round	50	0.960
6-5F-4	Rect.	54	0.953	12-2F-1	Square	58	0.905	21-5	Round	30	0.975
6-5F-2	Rect.	54	0.956	13-5	Rect.	77	0.917	21-4	Round	30	0.975
6-5F-1	Rect.	54	0.954	13-5F	Rect.	77	0.918	21-3	Round	30	0.970
7-5	Round	40	0.985	13-4	Rect.	77	0.906	21-2	Round	30	0.955
7-5F	Round	40	0.987	13-3	Rect.	77	0.893	22-5	Round	115	0.944
7-4	Round	40	0.976	13-3F	Rect.	77	0.918	22-4	Round	115	0.944
7-3	Round	40	0.968	13-2	Rect.	77	0.893	22-3	Round	115	0.945
7-3F	Round	40	0.978	13-2F	Rect.	77	0.922	22-2	Round	115	0.944
7-2	Round	40	0.958	14-5	Rect.	107	0.870	23-5	Square	115	0.890
7-2F	Round	40	0.962	14-5F	Rect.	107	0.878	23-4	Square	115	0.890
8-6	Square	40	0.940	14-5F-8	Rect.	107	0.882	23-3	Square	115	0.890
8-6F	Square	40	0.955	14-5F-4	Rect.	107	0.880	23-2	Square	115	0.884
8-5	Square	40	0.950	14-5F-2	Rect.	107	0.878				

* Major angle only for rectangular hoods.

are the values for flanged and unflanged duct ends (hood groups 18, 19 and 16-F-½R) which check very well with the usual values of 0.82 and 0.72, respectively. The results for hood group 21 (average about 0.97) agree fairly well with data on a similar hood given in reference 9 (0.944) but are considerably higher than results reported elsewhere,[5] where a value of about 0.875 is given.

The fact that most of the results obtained in this study are substantially higher than those in common use, caused considerable concern and led to check and recheck of the test procedure from all angles. As all attempts in this direction confirmed the results listed, the authors are at a loss to explain the lower values reported elsewhere. As one check on test accuracy, a con-

FIG. 2. SOME OF HOOD TYPES STUDIED. SMALL RATIO OF HOOD FACE AREA TO THROAT AREA

tracting jet orifice, borrowed from Case Institute of Technology, having, according to their studies, a coefficient of 0.985, was tested. Several tests on this orifice at a duct velocity of about 4000 fpm gave an average value of 0.989, which is a satisfactory check and seems to substantiate the results reported in this paper. It was suggested by one other worker[10] that the roughness of the taper to throat connection is extremely important and will influence the value of the coefficient a great deal; the smoother or rounder this joint, the higher will be the coefficient of entry. That the nature of this connection will have a significant influence on the energy loss at a hood seems apparent, yet a careful study of the construction of most of the hoods used in this study indicates the presence of two features which should place them in the high energy loss group. These features are: (1) the tapered section terminates inside the rolled edge of the throat section thereby presenting a straight and sharp edge at the point where the change in direction of air movement takes place and (2) at the rolled edge of some of the throat sections the pipe is drawn to form an internal bead and the inside diameter of which is significantly less than the pipe diameter (see hoods 4, 7 and 12 in Fig. 2). In spite of these apparent bad features in construction and the two different

types of construction discussed in the Appendix, the measured coefficients fall in line with each other, as will be demonstrated more clearly.

Effect of Distance from Throat on Results

As is shown in the Appendix, readings were taken at a number of distances downstream from the throat. This was done purposely to determine how far from the throat readings should be taken to get reliable results; that is, how far down the duct influence of the vena contracta would be felt sufficiently to give high or erratic results. This is a very important point in field work where the static reading is used to estimate the quantity of air being exhausted through a hood since the ductwork downstream from the hood is frequently so crooked, inaccessible or damaged as to make it imperative that the reading be taken very close to the hood.

The results indicate that a distance of one pipe diameter is adequate for most hoods but that a greater distance is desirable if the included angle of the

TABLE 2—UNFLANGED HOODS HAVING HIGHEST PRESSURE (LOWEST SUCTION) MORE THAN ONE PIPE DIAMETER DOWNSTREAM FROM HOOD THROAT

HOOD GROUP	SHAPE	MAJOR INCLUDED ANGLE DEG	SUCTION AT ONE PIPE DIAMETERS IN. WATER	LOWEST SUCTION IN. WATER	DISTANCE FROM HOOD THROAT IN PIPE DIAMETERS
4	Rectangular	109	1.378	1.356	1¾
5	Rectangular	134	1.361	1.349	1¾
13	Rectangular	77	1.262	1.258	1¾
15	Round	136	1.263	1.254	1¾
18	Round	0[a]	2.363	1.926	4½+
18-F	Round	180[b]	1.584	1.506	4½+

[a] Round unflanged open duct end.
[b] Flanged round duct end.

transformation section or hood is great. Of the 23 hood groups studied, having 8 in. diameter throats, only seven were found to have the lowest static suction reading more than one pipe diameter downstream from the throat. Pertinent data on six of these hood groups are summarized in Table 2. It is not apparent why hood group 13 is among this number. Other rectangular and square hoods having greater included angles have not given similar results. Furthermore, this same characteristic did not appear for all sizes of the hood group nor at all duct velocities. It is believed, therefore, that this hood group is among the lot by virtue of a small experimental error (note the small difference) or some unusual characteristic of the air stream. For open duct ends or flanged duct ends, however, the static reading should be taken not less than three pipe diameters (preferably 5) downstream from the throat. The results in Tables 1 and 2 indicate that even though the lowest static reading for some hoods with severe taper (over 100 deg included angle) exists more than one pipe diameter downstream from the throat, for all practical purposes one pipe diameter is satisfactory for included angles up to 180 deg (flanged duct ends). For included angles greater than 180 deg and for flanged

and unflanged duct ends, the reading should be taken at a point not less than 3 pipe diameters downstream from the throat; the suction beyond 3 pipe diameters decreases only very little.

Effect of Static Tap Location on Coefficient

The static suction close to the throat of a rectangular to round taper or rectangular hood is not the same at all locations around the throat, particularly

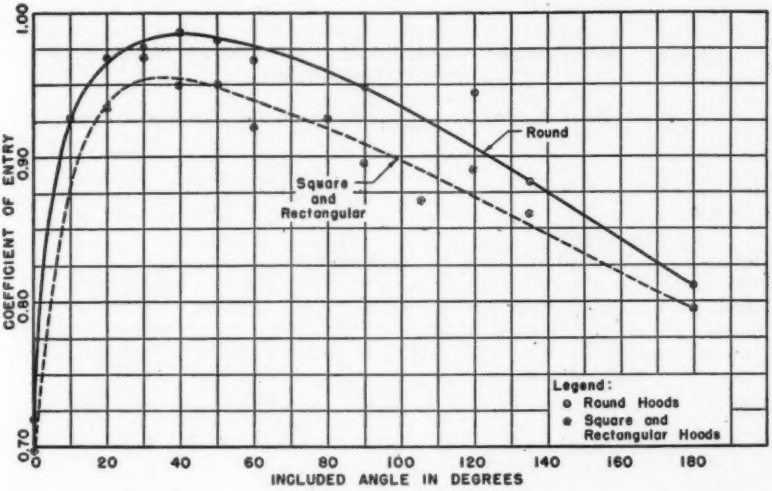

FIG. 3. EFFECT OF THE INCLUDED ANGLE IN UNFLANGED HOODS ON THE COEFFICIENT OF ENTRY

if the sides ratio of the hood is quite large. Readings taken near the throat are considerably higher in the area of the major included angle than they are in the area of the smaller angle. Results indicate, however, that even with a sides ratio of 4 to 1 and with major and minor included angles of 134 deg and 0 deg, respectively, the readings at one pipe diameter downstream from the throat vary only little at different locations around the duct as regards the relation to the different included angles. There is, of course, considerably more turbulence at this short distance from the throat with a hood of this type than with a round or square hood and suction readings should be taken at three or four points around the duct, if possible, to obtain the best results.

Effect of Duct Velocity on Coefficient of Entry

The coefficients of entry were determined at duct velocities of about 2000, 3000, 4000 and 5000 fpm on 70 different unflanged hoods. The total averages

of these coefficients at the different velocities are 0.925, 0.920, 0.922 and 0.922, respectively. Hence it is obvious that the duct velocity within the limits of 2000 to 5000 fpm has no significant influence on the coefficient of entry.

Effect of Hood Taper on Entrance Loss

The unflanged hoods studied included a large variety of different tapers. In Table 3 are summarized the coefficients for hoods of different included angles, all other variables having been kept constant, except as noted. It is apparent from the table that the best hood from the energy loss standpoint

TABLE 3—EFFECT OF THE INCLUDED ANGLE IN UNFLANGED HOODS ON THE COEFFICIENT OF ENTRY

(*Duct velocity about 4000 fpm* $\frac{A_1}{A_2}$ *about 5 except where otherwise stated*)

HOOD TYPE	INCLUDED ANGLE IN DEGREES												
	0[a]	10	20	30	40	50	60	80	90	105	120	135	180
	Coefficients of Entry												
Round	0.717	0.926[c]	0.967	0.975	0.985	0.980	0.966		0.948		0.944	0.883	0.810
Square	0.697				0.950		0.919		0.895		0.890		0.795
Rect.[b]			0.933[c]	0.969		0.950		0.927	0.870			0.860	

[a] Open end duct—same as 360 deg.
[b] Major angles only for rectangular hoods.
[c] $\frac{A_1}{A_2} = 2.5$

has an included angle of about 40 deg. This is true whether the hood is round, square or rectangular. The approximate coefficients for unflanged, round, square and rectangular hoods having throat diameters of 8 in. and face area to throat area ratios of 5, and for included angles of 0 to 180 deg are shown in Fig. 3. Of interest are the coefficients listed in Table 3 for hoods 16, 17, 20 and 21 (included angles of 90, 20, 50 and 30 deg respectively) since the construction of these hoods was such that the taper to throat section junction appeared to be considerably smoother than the other hoods (see Appendix). Yet the coefficients are not out of line with those of the other hoods. Both Table 3 and Fig. 3 indicate that in general the coefficients for square and rectangular hoods are in the order of 0.03 lower than for similar round hoods, this difference being greater in the middle section of the curve than at the extremities.

These data are at considerable variance with other information in the literature on this same subject. One source [6] suggests that the lowest energy loss is accomplished with a hood having an included angle of 13 deg. Another source [5] suggests that a hood having an included angle of 60 deg or more has

the same coefficient of entry as a flanged duct end. As indicated in Table 3, a 60 deg hood is an excellent one and the coefficient is in the order of 0.94 not 0.82 as for a flanged duct end. It must be noted, of course, that these values are for hoods having 8 in. diameter throats. For different size hoods the coefficients will differ as shown later.

Effect of Face Area to Throat Area Ratio on Entrance Loss

A total of 16 different hood groups were studied at four different hood face area to throat area ratios. These included round, square and rectangular

TABLE 4—EFFECT OF HOOD FACE AREA TO THROAT AREA RATIO ON COEFFICIENT OF ENTRY FOR UNFLANGED HOODS

(*Duct velocity 4000 fpm*)

HOOD GROUP	SHAPE	INCLUDED ANGLE[a] DEG	COEFFICIENTS FOR DIFFERENT AREA RATIOS						
			6	5	4	3	2	1.6	1.2
1	Round	10				0.926	0.922	0.888	0.823
2	Rectangular	17				0.933	0.925	0.903	
3	Square	90	0.901	0.895	0.894	0.898	0.887		
4	Rectangular	109		0.870	0.867	0.858	0.845		
5	Rectangular	134		0.860	0.862	0.855	0.858		
7	Round	40		0.985	0.976	0.968	0.958		
8	Square	40	0.940	0.950	0.944	0.942	0.933		
9	Rectangular	29		0.969	0.962	0.952	0.938		
10	Rectangular	81		0.936	0.939	0.926	0.912		
12	Square	58	0.910	0.919	0.922	0.916	0.895		
13	Rectangular	77		0.917	0.906	0.893	0.893		
15	Round	136		0.883	0.893	0.895	0.902		
16	Round	90		0.948	0.952	0.950	0.952	0.959	0.811
17	Round	21		0.967	0.963	0.960	0.946		
20	Round	50		0.980	0.978	0.973	0.960		
21	Round	30		0.975	0.975	0.970	0.955		
22	Round	115		0.944	0.944	0.945	0.944		
23	Square	115		0.890	0.890	0.890	0.884		

[a] Major angle only for rectangular hoods.

hoods. In addition a few hoods were studied at different ratios. The results of these tests are tabulated in Table 4.

It is apparent from these data that the face area to throat area ratio in the range of 5 to 2 has only a very small influence on the coefficient of entry; the coefficient decreasing as the ratio decreases for most hoods. The averages for all hood groups 3 to 23 inclusive are 0.931, 0.929, 0.924 and 0.916 for face area to throat area ratios of 5, 4, 3 and 2, respectively. It is interesting to note, however, that most of this reduction takes place among the hoods having a small degree of taper. Thus the average coefficients at the four different face area to throat area ratios for the seven hood groups having included angles of 90 deg or more are 0.899, 0.900, 0.899 and 0.896 as against the values given for all hood groups and 0.955, 0.952, 0.944 and 0.932, respectively, for the hoods having included angles of less than 90 deg. The ratio value of 2, however, appears to be the lower limit of the range in which the

coefficients are essentially constant. Of the three hoods studied at ratios below 2, only one remained essentially the same at a ratio of 1.6 and both hoods investigated at a ratio of 1.2 showed coefficients essentially the same as for flanged duct ends.

These data indicate that, contrary to common belief,[5] the hood face area to throat area ratio has very little effect on the coefficient of entry except for values less than 2. The reason for this is not apparent at the time of this writing. At first glance it would seem that the face area to throat area should

TABLE 5—EFFECT OF FLANGES ON COEFFICIENT OF ENTRY

(Duct velocity 4000 fpm, flange width about 12 in.)

HOOD NUMBER	SHAPE	INCLUDED ANGLE DEG	COEFFICIENT		HOOD NUMBER	SHAPE	INCLUDED ANGLE DEG	COEFFICIENT	
			Unflanged	Flanged				Unflanged	Flanged
1-2.5	Round	10	0.926	0.967	9-2	Rectangular	29	0.938	0.960
2-2.5	Rectangular	17	0.933	0.968	10-5	Rectangular	81	0.936	0.952
3-6	Square	90	0.901	0.908	10-3	Rectangular	81	0.926	0.943
3-3	Square	90	0.898	0.900	10-2	Rectangular	81	0.912	0.950
3-2	Square	90	0.887	0.902	11-5	Round	58	0.971	0.978
4-5	Rectangular	109	0.870	0.872	12-6	Square	58	0.910	0.923
4-3	Rectangular	109	0.858	0.873	13-5	Rectangular	77	0.917	0.918
4-2	Rectangular	109	0.945	0.879	13-3	Rectangular	77	0.893	0.918
5-5	Rectangular	134	0.860	0.883	13-2	Rectangular	77	0.893	0.922
5-3	Rectangular	134	0.855	0.886	14-5	Rectangular	107	0.870	0.878
5-2	Rectangular	134	0.858	0.892	15-5	Round	136	0.883	0.895
6-5	Rectangular	54	0.950	0.960	15-3	Round	136	0.895	0.891
7-5	Round	40	0.985	0.987	15-2	Round	136	0.902	0.885
7-3	Round	40	0.968	0.978	16-5	Round	90	0.948	0.955
7-2	Round	40	0.958	0.962	16-3	Round	90	0.950	0.955
8-6	Square	40	0.940	0.955	16-2	Round	90	0.952	0.947
8-3	Square	40	0.942	0.950	17-5	Round	21	0.967	0.983
8-2	Square	40	0.933	0.938	17-3	Round	21	0.960	0.975
9-5	Rectangular	29	0.969	0.981	17-2	Round	21	0.946	0.964
9-3	Rectangular	29	0.952	0.966					

have an influence on the coefficient since the loss at the hood face obviously is greater with a small than with a large area and it seems unlikely that this difference is made up by the greater length of the larger area hood. It is hoped that this problem may be investigated further to find the reason for these results.

Effect of Flanges on Coefficients

Of the tapered hoods investigated, 39 were studied with 12 in. wide flanges and without flanges. These results, summarized in Table 5, indicate a significant but unimportant increase in the coefficients with flanges. The average of all coefficients without flanges is 0.919 and with flanges is 0.933. All hoods except three (15-3, 15-2 and 16-2) showed increases with flanges. The exceptions are the round hoods with the greatest included angles and with small face area to throat area ratios. The reasons for these decreases are very

likely the same as those which caused the coefficients of these same hoods to increase as the face area to throat area decreased (see Table 4). Since the average increase is 0.014 and the median increase is 0.013, it may be concluded that in general the coefficient of a hood is increased about 0.015 by providing a 12 in. wide flange. For all practical purposes, however, this increase is too small to have significance.

Effect of Flange Width on the Coefficient

Flange widths of 12, 8, 4, 2 and 1 in. were used on some of the hoods investigated. The results of these tests are given in Table 6. It is obvious from these data that the flange width in the range studied has no significant effect upon the coefficient of entry. The average value for the coefficients of

TABLE 6—EFFECT OF FLANGE WIDTH ON COEFFICIENT OF ENTRY
(*Duct velocity 4000 fpm*)

HOOD NUMBER	SHAPE	INCLUDED ANGLE[a] DEG	COEFFICIENTS FOR DIFFERENT FLANGE WIDTHS				
			12 in.	8 in.	4 in.	2 in.	1 in.
4-3	Rectangular	109	0.873		0.876	0.880	
5-3	Rectangular	134	0.886	0.888	0.884	0.890	
5-2	Rectangular	134	0.892	0.895	0.892	0.894	0.888
6-5	Rectangular	54	0.960	0.949	0.953	0.956	0.954
8-3	Square	40	0.950	0.955	0.952	0.950	
8-2	Square	40	0.938	0.939	0.941	0.941	0.942
11-5	Round	58	0.978	0.979	0.974	0.973	0.974
12-6	Square	58	0.923	0.922	0.923	0.924	
14-5	Rectangular	107	0.878	0.882	0.880	0.878	0.883
18-F	Round	0	0.803[b]	0.811	0.814	0.818	0.835
19-F	Square	0	0.791	0.795	0.792	0.795	0.800

[a] Major angle only for rectangular hoods.
[b] 10 in. flange.

the seven hoods which were studied over the entire range of flange widths are 0.891, 0.893, 0.892, 0.894 and 0.897 for 12, 8, 4, 2 and 1 in. widths, respectively. On the basis of the data in Tables 5 and 6 it is logical to conclude that flanges have no significant effect on the coefficient of entry of hoods other than duct ends. Even a ½ in. wide flange (see hood 16-F-½R in Table 1) has about the same coefficient as wider flanges (see 18-F in Table 6).

Effect of Large Adjacent Surfaces on Coefficient

A few tests were run on different hoods with large flat surfaces adjacent to one, two, or three sides of the hoods, the surfaces being parallel to the center line of the exhaust duct. These conditions were intended to simulate those existing in industry when a hood is laid on, or located adjacent to, a table top and possibly one or two vertical partitions are located at the ends or sides of the hood to eliminate cross drafts or for other purposes. Even

though the number of tests was very small, the results indicate that such conditions have no important influence on the coefficient of entry.

Effect of Hood Throat Size on Coefficient

Since for similar hoods the ratios of the hood face perimeter and the throat perimeter to the hood face area and the throat area, respectively, decrease as the hood or throat size increases, it seemed logical that the coefficient of entry would be affected by the size for hoods of similar shape and proportions. It was believed that the coefficient would increase as the size increases and vice versa. Four comparable hoods of each 4 in. and 12 in. size were studied and the results are summarized in Table 7. These results are very interesting, since the larger size hoods did not show the anticipated lower energy losses and the coefficients of the smaller hoods were lower than expected. It is not

TABLE 7—EFFECT OF THROAT DIAMETER ON COEFFICIENT OF ENTRY

THROAT DIAMETER INCHES	COEFFICIENT FOR DIFFERENT INCLUDED ANGLES			
	0° or 360° (Open Duct)	40° $\left(\frac{A_1}{A_2} = 5\right)$	90° $\left(\frac{A_1}{A_2} = 5\right)$	180° (Flanged Duct)
4	0.702	0.926	0.919	0.793
8	0.717	0.985	0.948	0.810
12	0.695	0.962	0.952	0.802

clear why the coefficients of the larger hoods are lower than those of the hoods having 8 in. diameter throats. Variations of the order shown in the table for the 40 and 90 deg tapered hoods having 8 in. and 12 in. throats might be explained on the basis of minor differences in construction; a point already discussed. Other factors which may have contributed to these very small inconsistencies, particularly in the case of the plain and flanged duct ends, are: (1) inaccurate measurement of the inside pipe area, (2) roundness of open end, (3) thickness of metal and (4) type of construction. The inside diameters of the ducts were determined by measuring the diameter at three or four places and averaging these measurements. Since these values were found to be close to 8 in. and 12 in. respectively, these figures were used in the calculations and might easily have varied from the exact values sufficiently to have caused the small inconsistencies noted. The 8 in. duct was made of 16 gage metal and was definitely more nearly round than the 4 in. or 12 in. ducts. The greater thickness of the metal in the 8 in. duct, also, would tend to increase the coefficient since it served as a very narrow flange. This small difference may appear at first thought to be of no consequence yet actual tests on an 8 in. duct of 20 gage material gave a result 0.011 lower than that for the 16 gage duct although other variables probably contributed to this difference; also a ½ in. flange had a coefficient of 0.839 (hood 16-F-½R, Table 1). In addition the smooth seam of the 8 in. duct contrasted to the usual construction of the 12 in. duct, may have added a small amount to the higher coefficient

of the 8 in.. duct. These several factors may well account for the fact that the 8 in. coefficient (average 0.865) was slightly higher than the 12 in. coefficient (average 0.853) rather than vice versa. This difference however, is too small to have any practical significance. The difference between the coefficients for the 4 in. hoods (average 0.825) and the 8 in. hoods is sufficiently large to warrant consideration even in field work if accuracy of results is important. However, the greatest deviation of any single value in Table 7 from the average for all three sizes of similar hoods is less than 4 per cent. This is well within the range of permissible error for most field work.

CONCLUSION

The energy losses at the entrance to 175 different types of suction openings were determined and the coefficients of entry were calculated. Effects of the following factors on the coefficient of entry were studied: (1) Distance of suction measurement downstream from hood throat; (2) Duct velocity; (3) Degree of hood taper (included angle); (4) Hood shape; (5) Hood face area to throat area ratio; (6) Flanges; and (7) Size of hood throat.

On the basis of the data obtained in this study it may be concluded that:

1. Reliable results will be obtained if suction readings are taken one or more pipe diameters downstream from the throat for hoods having included angles up to 180 deg and three or more pipe diameters downstream from the throat for included angles of 180 deg (flanged pipe end) to 360 deg, and for 0 deg (plain pipe end).
2. Duct velocity in the range of 2000 to 5000 fpm has no significant effect on the coefficient of entry.
3. The degree of hood taper (included angle) is the most significant factor affecting the coefficient of entry (see Fig. 3). The coefficients vary from about 0.70 for 0 deg included angle (plain pipe end) to a maximum of about 0.98 for a round hood with an included angle of 40 deg.
4. The hood shape, whether round, square or rectangular, has a significant, though relatively unimportant, effect on the coefficient of entry. The coefficients for round hoods are in the order of 0.02 to 0.04 higher than for square and rectangular hoods. The differences are greater in the middle portion of the curve than at the ends (see Fig. 3).
5. The hood face area to throat area ratio in the range of five to two has no important effect upon the coefficient of entry. Below 2 this ratio becomes of concern, because the coefficient approaches that of a flanged duct end which it equals at a ratio of 1.2.
6. Flanges have no important influence on the coefficients of tapered hoods. They do, however, have a very important influence on the coefficients of plain duct ends (0 deg included angle).
7. The size of the hood throat has a significant, though not too important effect on the coefficient of entry for similar hoods. An 8 in. diameter throat gave higher results than 4 in. or 12 in. diameter throats, although the difference between the 8 in. and 12 in. results is not significant.
8. Entry coefficients accurate to within about plus or minus 5 per cent may be predicted if the hood shape and included angle are known. Curves for this purpose are shown in Fig. 4. These curves differ slightly from those in Fig. 3 since they have been modified to take into account the effect of the hood throat sizes or duct sizes in the range commonly encountered in local exhaust systems.

Most of the results obtained in this study differ considerably from many of those reported in the literature and in common use today. With the exception of plain duct ends and flanged duct ends the results shown are higher than

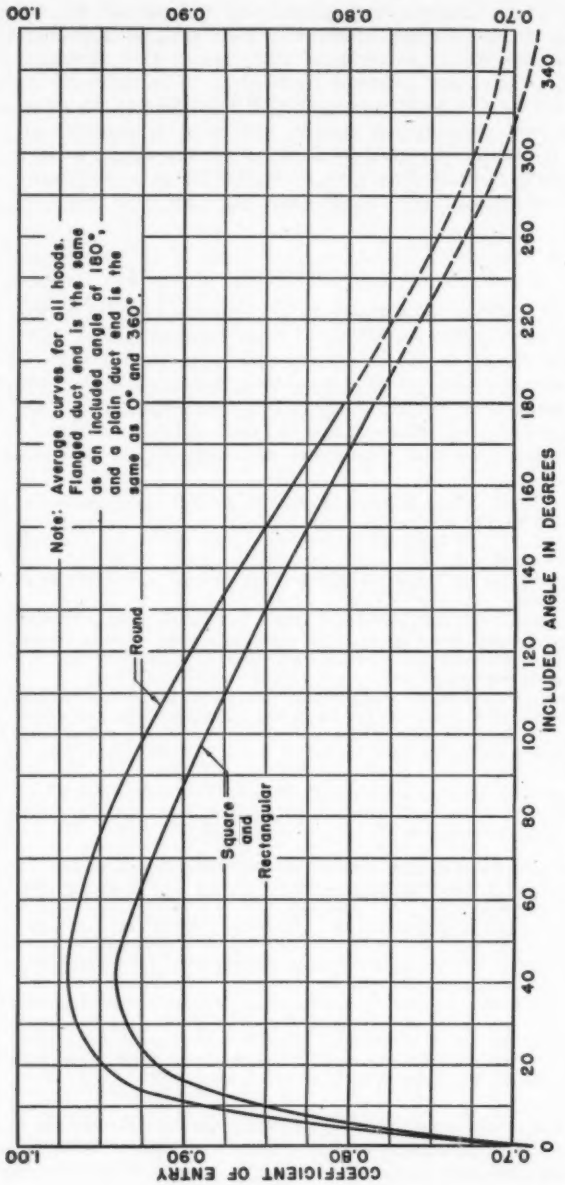

FIG. 4. COEFFICIENTS OF ENTRY FOR HOODS UNDER SUCTION

most of those cited in the literature. All attempts to find the reason for these differences were unsuccessful.

When considering the data and statements in this paper, it must be borne in mind that reference is made only to the coefficient of entrance, not to the relative merits of one hood to another with regards to its performance. For example, while a flange does not improve a hood in regard to entrance loss, it improves the air flow pattern in front of the hood a great deal; the wider the flange the better, in this respect. Likewise, the hood face area, while unimportant from the viewpoint of entrance losses, has a decided influence on the air flow pattern in front of the hood. Consequently, the statements in this paper must not be construed as implying that one hood is no better than another in regard to the air flow pattern produced by it.

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APPENDIX

Experimental Procedure

Most of the hoods studied had 8 in. diameter throats since the available experimental setup had 8 in. diameter ductwork and metering devices. The test arrangement with a round hood attached is shown in Fig. 1. The hood throats were slightly less than 8 in. in diameter to accomplish a snug inside lap fit. An air seal between the hood throats and duct was accomplished by means of adhesive tape as shown in the figure. The metering device was a 6.00 in. sharp-edged orifice located about 26 ft (39 duct diameters) downstream from the hood. The computed calibration curve

for the orifice checked very closely with rates of air flow determined by pitot tube traverses in the duct. The variations of the computed rate of air flow from that determined by pitot tube studies by different individuals at different rates of air flow and on different days were all less than plus or minus 3 per cent.

The end of the duct to which the hoods were attached for study was a carefully made, butt seamed, almost truly round, 24 in. long, section of pipe of 16 gage material. The ¼ in. thick flange which was welded to the pipe at the downstream end served also to maintain the shape of the pipe almost truly round under all conditions. Static taps having 1/32 in. holes through the pipe wall were located in a straight line and three inches on center as shown in Fig. 5. The end taps were each 3 in. from the ends of the pipe. In addition to the single taps at the 3 and 18 in. locations three other taps were located around the pipe at 90 deg intervals (see Figs. 1 and 5) permitting average static readings around the pipe at these two locations. An addi-

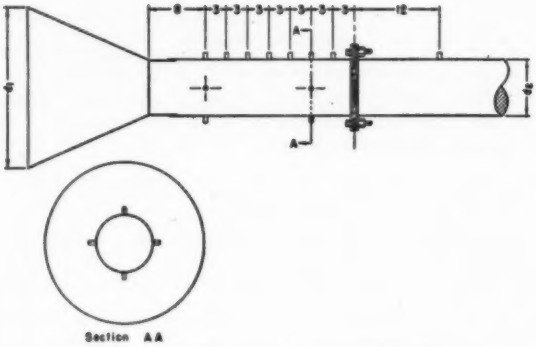

FIG. 5. DUCT SECTION SHOWING STATIC TAPS

tional tap was located 12 in. beyond the flange or 36 in. from the open end of the pipe as shown in the figures. The length of the throat section of all hoods having 8 in. diameter throats was 6 in. so that by overlapping the pipe and throat 1 in. the first tap was located one pipe diameter downstream from the throat and successive taps were located at 3 in. intervals up to 26 in. from the throat and the last tap at 41 in. from the throat.

The pressure drop across the orifice was measured by means of a vertical water manometer and the static suction readings were made with carefully calibrated inclined water manometers (see Fig. 1) in most instances. Vertical manometers were used from time to time as a double-check on the inclined gages. As a general rule, static readings on all hoods were made at one pipe diameter and three pipe diameters (actually 23 in.) from the throat at pipe velocities of about 2000, 3000 and 5000 fpm, and readings were made at all taps with a pipe velocity of 4000 fpm, a common duct velocity in industrial local exhaust systems. A large number of the readings at the 8 in. and 23 in. stations were obtained at both the single tap in the line of taps and at the interconnected taps spaced 90 deg apart. The purpose of this was to be certain that the single tap reading was correct, particularly with rectangular and square hoods where the static suction close to the throat is not the same at all points around the duct.

Hoods Studied

Data were collected for a variety of hoods most of which had 8 in. diameter throats facilitating convenient connection to the 8 in. duct diameter of the test equipment.

A few hoods of different size were studied also. Suitable transformation sections and straight duct sections were used to permit attaching the different throat size hoods to the test equipment and to permit obtaining satisfactory air flow and suction readings. A 4.00 in. diameter orifice was substituted for the 6.00 in. one for the smaller hoods.

The hoods having 8 in. diameter throats were divided between round, square and rectangular; the rectangular group being further varied in regard to the ratio of the width to length of the hood face dimensions. Each shape group contained a variety of hoods as regards the rate of divergence of the taper, or the included angle. The ratio of the hood face area to the throat area was varied from about 5 (6½ for square hoods) to 2 (less for a few hoods) by cutting down the length of the hoods with shears.

The hoods were all made by the same sheet metal contractor. Twenty gage material was used and the *usual industrial construction* was specified. The throat

FIG. 6. SOME OF HOOD TYPES STUDIED. LARGE RATIO OF HOOD FACE AREA TO THROAT AREA

sections were lap jointed and riveted. All the hoods were not ordered at one time; they were made on at least four different orders at different times over a period of about three months. It is not likely that they were all made by the same sheet metal worker since two distinct types of construction are apparent in Figs. 2 and 6. Hoods 16, 17, 20 and 21 present smoother taper to throat connection than do the others, yet it is the opinion of the authors that these hoods were representative of the type of construction usually encountered in industrial installations.

Seventeen of the hoods studied are shown in Fig. 6. This group shows the hoods in their original sizes, the face area to throat area being about 5 except for the square hoods which was about 6½. In Fig. 2 are shown the hoods after they had been reduced to the smallest face area to throat area ratio, about 2 for all except Nos. 1, 2 and 16 which were cut down even farther. The intervening ratio hoods (4 and 3) are not shown. Flanging of the hoods was accomplished by attaching snugly fitting one piece sections of light weight wall board over the hoods so that they were flush with the edges of the hoods at the face.

Pertinent data on all the unflanged round hoods are given in Table 8. A total of 44 hoods of this type were studied resulting in comparable data for ten different

TABLE 8—UNFLANGED ROUND HOODS STUDIED

(Dimensions and data)

HOOD NUMBER	(REFER TO FIG. 7 FOR LETTER DESIGNATIONS)						
	d_1 In.	d_2 In.	A_1 Sq In.	A_2 Sq In.	$\frac{A_1}{A_2}$	L In.	Included Angle Deg
1-2.5	12¾	8	128	50	2.6	27½	10
1-2	11	8	95	50	1.9	18	10
1-1.6	10	8	79	50	1.6	11½	10
1-1.4	9½	8	71	50	1.4	8¾	10
1-1.2	8¾	8	60	50	1.2	4¼	10
7-5	18	8	254	50	5.1	13¾	40½
7-4	16	8	202	50	4.0	11	40½
7-3	13¾	8	149	50	3.0	8	40½
7-2	11¼	8	99	50	2.0	4½	40½
11-5	18	8	254	50	5.1	9	58
15-5	18	8	254	50	5.1	2	136½
15-4	16	8	202	50	4.0	1½	136½
15-3	13¾	8	149	50	3.0	1¼	136½
15-2	11¼	8	99	50	2.0	½	136½
16-5	18	8	254	50	5.1	5	90
16-4	16	8	202	50	4.0	4	90
16-3	13¾	8	149	50	3.0	3	90
16-2	11¼	8	99	50	2.0	1½	90
16-1.6	10¼	8	82	50	1.6	1	90
16-1.4	9½	8	71	50	1.4	¾	90
16-1.2	8¾	8	60	50	1.2	½	90
17-5	18	8	254	50	5.1	27	21
17-4	16	8	202	50	4.0	21¼	21
17-3	13¾	8	149	50	3.0	15¾	21
17-2	11¼	8	99	50	2.0	8¾	21
18-1	8	8	50	50	1.0	—	0
20-5	17¾	8	247	50	4.9	11	50
20-4	16	8	202	50	4.0	9	50
20-3	14	8	154	50	3.1	6¾	50
20-2	11¼	8	99	50	2.0	3¾	50
21-5	18	8	254	50	5.1	19	30
21-4	16	8	202	50	4.0	15¼	30
21-3	14	8	154	50	3.1	11½	30
21-2	11¼	8	99	50	2.0	6¼	30
22-5	18	8	254	50	5.1	3¼	115
22-4	16	8	202	50	4.0	2½	115
22-3	13¾	8	149	50	3.0	2	115
22-2	11½	8	104	50	2.1	1	115
24-5	9	4	64	13	4.9	6½	40
25-5	9	4	64	13	4.9	2½	90
26-1	4	4	13	13	1.0		0
27-5	27	12	572	113	5.1	7½	90
28-5	27	12	572	113	5.1	21	40
29-1	12	12	113	113	1.0		0

rates of taper and for at least four different face area to throat area ratios on eight different hoods. As is shown in the table, the unflanged round group of hoods studied had tapers (included angles) of about 0, 10, 20, 30, 40, 50, 60, 90, 115 and 135 deg. A hood having a taper of 180 deg is really a flanged duct end and hoods of this type were included in the study (see Table 11). Greater tapers than 180 deg were not studied because they are seldom if ever used in industry. Furthermore, the coefficient

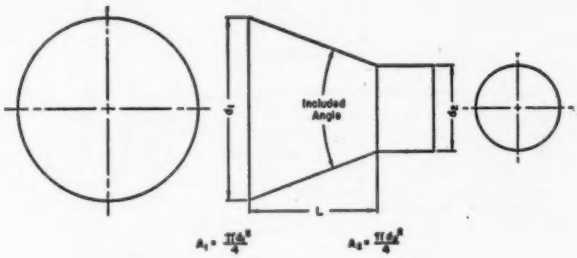

FIG. 7. ROUND UNFLANGED HOODS

of entry can be estimated sufficiently accurate for all practical purposes from the data for 180 and 0 deg (same as 360 deg) as indicated in Fig. 4. For a better understanding of the data given in Table 8, refer to Fig. 7. The hood numbers as used in Table 8 and other tables are significant in that the key or first number is the number assigned to the hood group as shown in Figs. 2 and 6 while the number following the dash is the approximate hood face area to throat area ratio. Thus 16-3 means hood No. 16 which has a face area to throat area ratio of about 3. Hood 16 shown in Fig. 6 is actually 16-5 while hood 16 shown in Fig. 2 is 16-1.2.

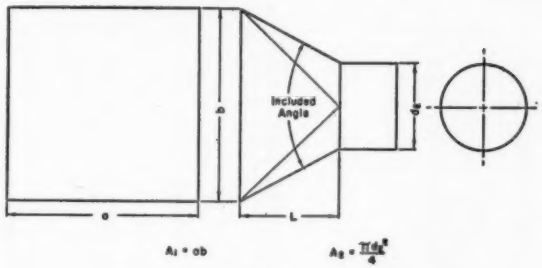

FIG. 8. SQUARE UNFLANGED HOODS

Four different unflanged, square hood groups and one individual hood were studied. Pertinent information on these hoods is given in Table 9 and Fig. 8. The included angles of this group were approximately 0, 40, 60, 90 and 115 deg. The face area to throat area ratios were the same as for round hoods except that for three hoods a ratio of about 6.5 also was studied. The numbering system for the hoods shown in Table 9 is the same as that for the hoods in Table 8.

Six different unflanged rectangular hood groups and two individual unflanged rectangular hoods were studied. Detailed information on these hoods is given in Table 10 and Fig. 9. Since in rectangular hoods the sides taper at two different rates, the included angles of both are given, the larger being the major and the

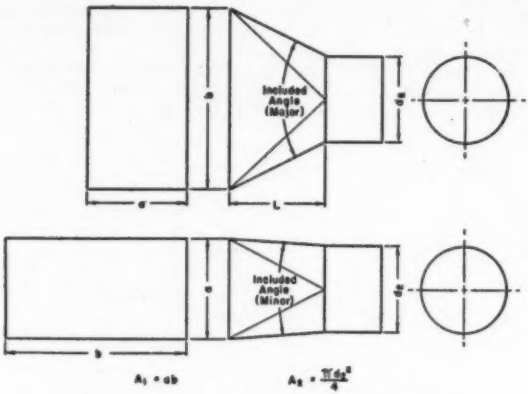

FIG. 9. RECTANGULAR UNFLANGED HOODS

smaller being the minor angle. The side ratios of the hood faces vary considerably even within the same hood group since the smaller hood sizes were made by cutting down the larger hoods. This procedure, while convenient from the viewpoint of cost, was not entirely desirable since it made difficult a good study of the effect, if any, of the sides ratio on the coefficient of entry.

TABLE 9—UNFLANGED SQUARE HOODS STUDIED

(*Dimensions and data*)

HOOD NUMBER	(REFER TO FIG. 8 FOR LETTER DESIGNATIONS)							
	a In.	b In.	d_2 In.	A_1 ($a \times b$) Sq In.	A_2 Sq In.	$\frac{A_1}{A_2}$	L In.	Included Angle Deg
3-6	18	18	8	324	50	6.5	5	90
3-5	16	16	8	256	50	5.1	4	90
3-4	14	14	8	196	50	3.9	3	90
3-3	12	12	8	144	50	2.9	2	90
3-2	10	10	8	100	50	2.0	1	90
8-6	18	18	8	324	50	6.5	13¾	40½
8-5	16½	16½	8	272	50	5.4	11¾	40½
8-4	14	14	8	196	50	3.9	8¼	40½
8-3	12	12	8	144	50	2.9	5½	40½
8-2	10	10	8	100	50	2.0	2¾	40½
12-6	18	18	8	324	50	6.5	9	58
12-5	16¼	16¼	8	264	50	5.3	7½	58
12-4	14½	14½	8	210	50	4.2	5¾	58
12-3	11½	11½	8	132	50	2.6	3¼	58
12-2	10	10	8	100	50	2.0	1¾	58
19-1	8	8	8[a]	64	64	1.0		0
23-5	16	16	8	256	50	5.1	2½	115
23-4	14	14	8	196	50	3.9	2	115
23-3	12	12	8	144	50	2.9	1¼	115
23-2	10	10	8	100	50	2.0	½	115

[a]Square.

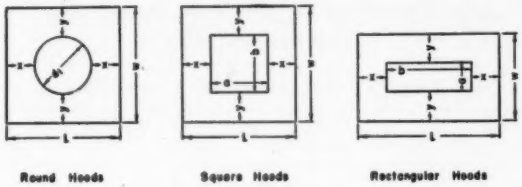

FIG. 10. FLANGED HOODS

It will be noted that in Tables 8, 9 and 10 most hood throat diameters (d_3) are given as 8 in. This is not strictly true since all 8 in. throat hoods excepting 16 and 17 had throats small enough to fit inside the 8 in. duct to which they were attached. Consequently, their effective inside diameters were slightly less than 8 in. They were, however, close enough to this dimension to permit ignoring it without introducing any significant error in the results. Hoods 16 and 17 on the other hand had 8 in. diameter throats and were, therefore, butted to the test duct and held in place with adhesive tape for study.

A total of 85 flanged hoods involving about 17 hood groups and two individual hoods were studied. Pertinent data on these hoods and flanges are given in Table 11

TABLE 10—UNFLANGED RECTANGULAR HOODS STUDIED

(Dimensions and data)

HOOD NUMBER	(REFER TO FIG. 9 FOR LETTER DESIGNATIONS)									
	a In.	b In.	$\frac{b}{a}$	d_3 In.	$\frac{A_1}{(a \times b)}$ Sq In.	A_2 Sq In.	$\frac{A_1}{A_2}$	L In.	Included Angle Deg	
									Major	Minor
2-2.5	8	16	2.0	8	128	50	2.6	27	17	0
2-2	8	12	1.5	8	96	50	1.9	13½	17	0
2-1.6	8	10	1.3	8	80	50	1.6	6¾	17	0
4-5	11	22	2.0	8	242	50	4.8	5	109	33½
4-4	10¼	19½	1.9	8	200	50	4.0	4	109	33½
4-3	9½	15½	1.6	8	147	50	2.9	2¾	109	33½
4-2	8¾	11¼	1.3	8	98	50	2.0	1¼	109	33½
5-5	8	31½	3.9	8	252	50	5.0	5	134	0
5-4	8	24¼	3.0	8	194	50	3.9	3¾	134	0
5-3	8	19	2.4	8	152	50	3.0	2½	134	0
5-2	8	12	1.5	8	96	50	1.9	1	134	0
6-5	11	22	2.0	8	242	50	4.8	13¾	54	12½
9-5	11	22	2.0	8	242	50	4.8	27¼	29	6½
9-4	10	18¼	1.8	8	183	50	3.7	20	29	6½
9-3	9¼	15	1.6	8	139	50	2.8	13½	29	6½
9-2	8½	11¼	1.3	8	96	50	1.9	6¼	29	6½
10-5	8	31½	3.9	8	252	50	5.0	13¾	81	0
10-4	8	23¾	3.0	8	190	50	3.8	9¼	81	0
10-3	8	18	2.3	8	144	50	2.9	5¾	81	0
10-2	8	12	1.5	8	96	50	1.9	2¼	81	0
13-5	11	22	2.0	8	242	50	4.8	8¾	77½	20
13-4	10¼	19½	1.9	8	200	50	4.0	7¼	77½	20
13-3	9¼	15	1.6	8	139	50	2.8	4¼	77½	20
13-2	8½	11¼	1.3	8	96	50	1.9	1¾	77½	20
14-5	8	31½	3.9	8	252	50	5.0	8¾	107	0

TABLE 11—FLANGED HOODS STUDIED

(Dimensions and data)

HOOD NUMBER	(REFER TO FIGS. 7, 8, 9 AND 10 FOR LETTER DESIGNATIONS)									
	d_1 In.	a In.	b In.	A_1 Sq In.	L In.	w In.	x In.	y In.	A_2 Sq In.	$\frac{A_2}{A_1}$
1-2.5F	12¾			128	42	42	14½	14½	1764	13.8
2-2.5F		8	16	128	46	35	15	13½	1610	12.6
3-6F		18	18	324	42	42	12	12	1764	5.4
3-3F		12	12	144	36	36	12	12	1296	9.0
3-2F		10	10	100	34	34	12	12	1156	11.6
4-5F		11	22	242	46	35	12	12	1610	6.7
4-3F		9½	15½	147	39½	33½	12	12	1324	9.0
4-3F-4		9½	15½	147	23½	17½	4	4	411	2.8
4-3F-2		9½	15½	147	19½	13½	2	2	263	1.8
4-2F		8¾	11¼	98	35¼	32¼	12	12	1155	11.8
5-5F		8	31½	252	56	32	12¼	12	1792	7.1
5-3F		8	19	152	43	32	12	12	1376	9.1
5-3F-8		8	19	152	35	24	8	8	840	5.5
5-3F-4		8	19	152	27	16	4	4	432	2.8
5-3F-2		8	19	152	23	12	2	2	276	1.8
5-2F		8	12	96	36	32	12	12	1152	12.0
5-2F-8		8	12	96	28	24	8	8	672	7.0
5-2F-4		8	12	96	20	16	4	4	320	3.3
5-2F-2		8	12	96	16	12	2	2	192	2.0
5-2F-1		8	12	96	14	10	1	1	140	1.5
6-5F		11	22	242	46	35	12	12	1610	6.7
6-5F-8		11	22	242	38	27	8	8	1025	4.2
6-5F-4		11	22	242	30	19	4	4	570	2.4
6-5F-2		11	22	242	26	15	2	2	390	1.6
6-5F-1		11	22	242	24	13	1	1	312	1.3
7-5F	18			254	42	42	12	12	1764	6.9
7-3F	13¾			149	37¾	37¾	12	12	1425	9.6
7-2F	11¼			99	35¼	35¼	12	12	1243	12.5
8-6F		18	18	324	42	42	12	12	1764	5.4
8-3F		12	12	144	36	36	12	12	1296	9.0
8-3F-8		12	12	144	28	28	8	8	784	5.4
8-3F-4		12	12	144	20	20	4	4	400	2.8
8-3F-2		12	12	144	16	16	2	2	256	1.8
8-2F		10	10	100	34	34	12	12	1156	11.6
8-2F-8		10	10	100	26	26	8	8	676	6.8
8-2F-4		10	10	100	18	18	4	4	324	3.2
8-2F-2		10	10	100	14	14	2	2	196	2.0
8-2F-1		10	10	100	12	12	1	1	144	1.4
9-5F		11	22	242	46	35	12	12	1610	6.7
9-3F		9¼	15	139	39	33¼	12	12	1297	9.3
9-2F		8½	11¼	96	35¼	32½	12	12	1146	11.9
10-5F		8	31½	252	56	32	12¼	12	1792	7.1

[a] Round flanges.

and Figs. 7, 8, 9 and 10. The numbering system is again significant in that the usual numbers for unflanged hoods are followed by the letter *F* for all flanged hoods and, unless otherwise indicated, the flange width was about 12 in. For other flange widths the actual width in inches follows the letter *F* and is separated therefrom by a dash. Most flanges on round hoods were square as indicated in Fig. 10, but a few of the narrow flanges on round hoods were circular following the outline of the

TABLE 11—FLANGED HOODS STUDIED—(*Concluded*)

(*Dimensions and data*)

HOOD NUMBER	(REFER TO FIGS. 7, 8, 9 AND 10 FOR LETTER DESIGNATIONS)									
	d_1 In.	a In.	b In.	A_1 Sq In.	L In.	w In.	x In.	y In.	A_2 Sq In.	$\frac{A_2}{A_1}$
10-3F		8	18	144	42	32	12	12	1344	9.3
10-2F		8	12	96	36	32	12	12	1152	12.0
11-5F	18			254	42	42	12	12	1764	6.9
11-5F-8	18			254	34	34	8	8	1156	4.5
11-5F-4	18			254	26	26	4	4	676	2.7
11-5F-2	18			254	22	22	2	2	484	1.9
11-5F-1	18			254	20	20	1	1	400	1.6
11-5F-1R	18			254	20[a]	20[a]	1[a]	1[a]	314	1.2
12-6F		18	18	324	42	42	12	12	1764	5.4
12-6F-8		18	18	324	34	34	8	8	1156	3.6
12-6F-4		18	18	324	26	26	4	4	676	2.1
12-6F-2		18	18	324	22	22	2	2	484	1.5
12-2F-2		10	10	100	14	14	2	2	196	2.0
12-2F-1		10	10	100	12	12	1	1	144	1.4
13-5F		11	22	242	46	35	12	12	1610	6.7
13-3F		9¼	15	139	39	33¼	12	12	1297	9.3
13-2F		8½	11¼	96	35¼	32½	12	12	1146	11.9
14-5F		8	31½	252	56	32	12	12	1792	7.1
14-5F-8		8	31½	252	47½	24	8	8	1140	4.5
14-5F-4		8	31½	252	39½	16	4	4	632	2.5
14-5F-2		8	31½	252	35½	12	2	2	426	1.7
14-5F-1		8	31½	252	33½	10	1	1	335	1.3
15-5F	18			254	42	42	12	12	1764	6.9
15-3F	13¾			149	37¾	37¾	12	12	1425	9.6
15-2F	11¼			99	35¼	35¼	12	12	1243	12.5
16-5F	18			254	42	42	12	12	1764	6.9
16-3F	13¾			149	37¾	37¾	12	12	1425	9.6
16-2F	11¼			99	35¼	35¼	12	12	1243	12.5
16-F-½R	8			50	9[a]	9[a]	½[a]	½[a]	64	1.3
17-5F	18			254	42	42	12	12	1764	6.9
17-3F	13¾			149	37¾	37¾	12	12	1425	9.6
17-2F	11¼			99	35¼	35¼	12	12	1243	12.5
18-F-10	8			50	28	28	10	10	784	15.7
18-F-8	8			50	24	24	8	8	576	11.5
18-F-4	8			50	16	16	4	4	256	5.1
18-F-2	8			50	12	12	2	2	144	2.9
18-F-1	8			50	10	10	1	1	100	2.0
18-F-1R	8			50	10[a]	10[a]	1[a]	1[a]	78	1.6
19-F	8			50	28	28	10	10	784	15.7
19-F-8	8			50	24	24	8	8	576	11.5
19-F-4	8			50	16	16	4	4	256	5.1
19-F-2	8			50	12	12	2	2	144	2.9
19-F-1	8			50	10	10	1	1	100	2.0

[a] Round flanges.

hood edge. These are identified by the letter R following the flange width. Thus 11-5F-1R signifies hood No. 11 with a face area to throat area ratio of 5, a 1 in. wide flange which was round in shape.

Of interest to some readers are the data shown in Figs. 11 and 12 which give a fair idea of the air flow conditions obtaining in hoods of different rates of convergence and in similar hoods having different face areas.

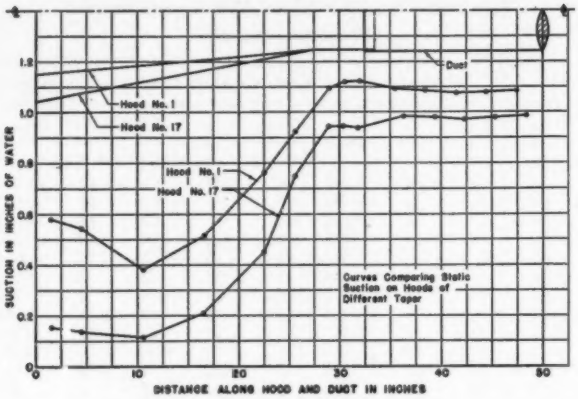
Duct
Hood No. 1
Hood No. 17
Hood No. 1
Hood No. 17
Curves Comparing Static Suction on Hoods of Different Taper
SUCTION IN INCHES OF WATER
DISTANCE ALONG HOOD AND DUCT IN INCHES

FIG. 11. ENERGY LOSS IN HOODS. COMPARISON OF STATIC SUCTION ON HOODS OF DIFFERENT TAPER

In Fig. 11 are shown the suction readings taken at various points on hoods 1 and 17, having included angles of 10 and 20 deg respectively and having face area to throat area ratios of 2.5 and 5, respectively. The curves are self-explanatory and show the influence of the vena contracta at the hood entrance, the static regain up to about 10 in. from the face, the influence of the increasing velocity as the air passes through smaller and smaller areas and finally the influence of the change in direction at the throat. The difference in the nature of the suction readings at the throat for the two hoods was caused by a difference in the throat diameters; hood No. 1 having a smaller diameter than the duct lying downstream thereof, whereas hood No. 17

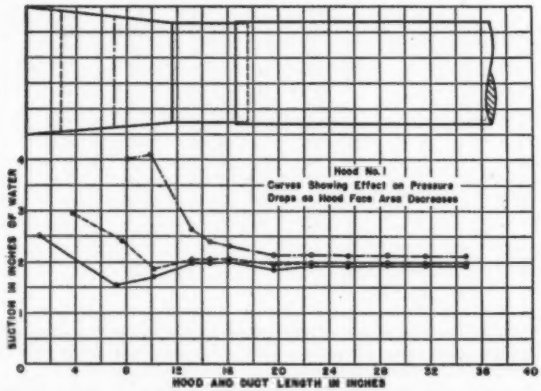
Hood No. 1
Curves Showing Effect on Pressure Drops as Hood Face Area Decreases
SUCTION IN INCHES OF WATER
HOOD AND DUCT LENGTH IN INCHES

FIG. 12. ENERGY LOSS IN HOODS. EFFECT OF HOOD FACE AREA ON PRESSURE DROP

had an effective inside diameter slightly larger than that of the downstream duct and was connected thereto by means of a butt joint as pointed out earlier.

In Fig. 12 are plotted a few suction readings taken on the same hood but in three different stages of face area reduction; face area to throat area ratios of 1.6, 1.4 and 1.2. Thus the top curve shows the readings when the hood was cut down to the size indicated by the *broken line and dash* line, the middle curve shows the readings when the hood was of the size shown by the *broken line* and the bottom curve shows the readings taken when the hood was the full size as shown in the figure. It is obvious of course, that the suction readings would not follow the straight lines joining the points where readings were taken. Data should have been collected at many more points to permit drawing accurate curves between them. The points of interest which are demonstrated by the data in the figure are the rate of increase in suction near the face as the area decreases and the following static regain so that the readings in the duct downstream of the hood are of the same order of magnitude. Had the face area to throat area all been above the critical value of about 2, the three curves would converge into one at, or slightly before, one pipe diameter downstream from the throat.

DISCUSSION

W. C. L. HEMEON, Pittsburgh, Pa. (WRITTEN): A few years ago we prepared for the Industrial Hygiene Foundation a bulletin describing methods for measuring air flow in industrial exhaust systems, the most important portion of which dealt with the static suction technique.[5*] We were inspired in this procedure by the excellent original exposition by John L. Alden.[1]

We desired to extend the data on entry coefficients beyond those given by Alden and concluded that coefficients could be derived by calculations based on hydrodynamic theory and shock loss data available in the literature. The coefficients that were presented were prepared by combining available data for the energy loss at the face of the hood with similar data applicable to the junction of the taper and the pipe, *i.e.*, throat.

Specifically, the coefficients resulted from an assumption that the loss at the face of an unflanged hood is 0.93 face velocity heads and that the loss at the throat is a function of the included angle. McElroy[7] cites values for the latter which we employed in part to derive our coefficients; the resulting coefficients agree well with experimental values reported by DallaValle[4] and also with values from Gibson and Castel (quoted by Baumeister).

The extensive test data obtained by the authors of this paper appear to be in disagreement with other data including ours. The following analysis of their data indicates that it is new data that need not be in agreement with others.

The authors, in their discussion, refer to included angle and to area ratios but have overlooked the taper length, a dimension of great importance. In cutting down the original hoods to vary the area ratio, they also changed the taper length and thus introduced two variables instead of one.

Nearly all the hoods studied by the authors are short. Only six have a taper length L exceeding one face diameter D. Our data were intended to be limited to hoods where the L/D is not less than 1 to 1½. Their data, therefore, extend the field of knowledge to shorter hoods, many of which are encountered in industry.

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* Exponent numerals in Discussion also refer to Bibliography.

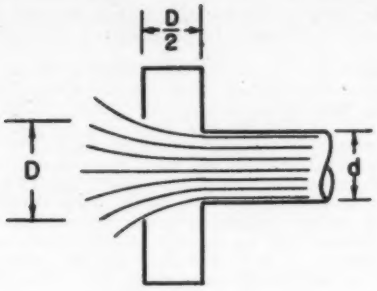

FIG. A. THEORETICAL SUCTION OPENING DESIGNED FOR COINCIDENCE OF THROAT WITH VENA CONTRACTA $(D/d)^2 = 0.62$

The importance of the L/D ratio is in its effect on the vena contracta just inside the hood face, the principle of which is illustrated in the special shape shown in Fig. A. The face and duct areas are related in the ratio of 1 to 0.62, which is the ratio of areas of a sharp edge orifice and its vena contracta. The distance between orifice and duct approximates the natural distance between orifice plate and the vena contracta. For such an opening we would expect a coefficient close to 0.98 or 0.99 (provided, further, that turbulence in the connecting chamber is insignificant).

In our view the short taper hoods described in the present study are identical in principle with the shape as suggested further in Fig. B. The contracted stream of air at the entrance, in both cases is directed into the duct and a major part of its energy is employed usefully, rather than being lost in shock inside the taper as would be expected in a taper long enough to complete the process of expansion.

In this analysis we have considered only the unflanged conical hoods, but the methods are equally applicable to the other shapes.

We have calculated the theoretical coefficient for hoods of different area ratios where no loss is assumed at entrance to the throat. Hoods with small included

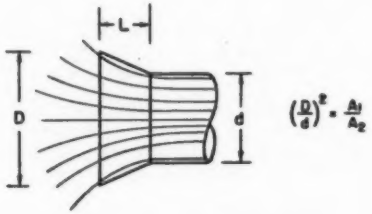

FIG. B. HOOD WITH INCLUDED ANGLE OF 45 DEG WHERE L/D IS $0.4D$, OR LESS, SO VENA CONTRACTA COINCIDES WITH DUCT. ALSO A_1/A_3 IS CLOSE TO 1.85, THE NATURAL RATIO OF A_1/A_{vena}

angle (*e.g.* 10 deg) closely approach this condition. The values, plotted in Fig. C, are obtained from the relation

$$C_e = \frac{1}{1 + 0.93\left(\frac{A_2}{A_1}\right)^2}$$

Also plotted thereon as circles are the authors' values for the coefficients of hoods with long tapers. They are in good agreement with our theoretical calculated values. The values of L/D range from 0.5 to 1.0. Values that appear as crosses and are

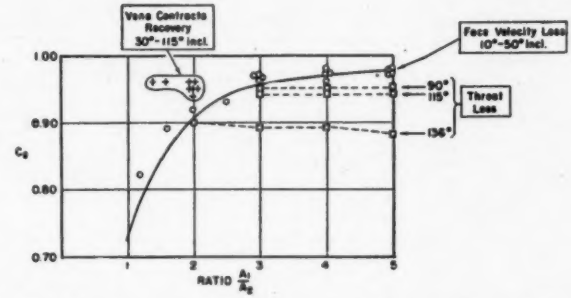

FIG. C. COEFFICIENTS ARRANGED IN 3 GROUPS—THOSE COINCIDING WITH THEORETICAL CURVE—THOSE WHERE VENA CONTRACTA VELOCITY RECOVERY IS HIGH—THOSE WHERE THROAT LOSS IS CONTROLLING

well above predicted values are the subject of special attention in the following discussion.

At what point relative to the face of an unflanged hood is the vena contracta located, and what is the coefficient of contraction? These facts may be deduced from data given in the appendix showing static pressures at points along the taper. They indicate clearly that the vena contracta is located inside the face at a distance about 0.1 the face diameter and maintains itself up to about ⅓ of the face diameter. It is also clear that recovery is complete at 0.6 to 0.8 D, average about ⅔ D.

A second fact of importance can be derived from these same data. From the static pressure of 0.58 in. just inside the face of hood 1, and related data, we can calculate that the coefficient of contraction is about 0.54, which may be compared with 0.62 for the sharp-edge orifice in a plate. Thus the area ratio A_1/A_{vena} is 1.85.

The significance of these factors may be summarized as follows:

1. If a hood has dimensions such that the length is more than about 0.6 D, enough space is provided so that expansion from the vena contracta effect is largely completed within the taper and, therefore, best agreement would be expected with theoretical calculations.

2. If the taper length is 1/10 D to 1/3 D, the throat will coincide with the position of the vena contracta, and its velocity energy will be utilized in the duct to result in a lower overall energy loss and a higher coefficient of entry.

3. If the A_1A_2 ratio of the hood coincides approximately with the ratio 1.85, that is the natural contraction of the entering stream, utilization of the velocity energy of the vena contracta would be greatest, provided that the conditions of (2) are also favorable, *i.e.*, a taper length of 1/10 to 1/3 D.

If we now examine the values that are unexpectedly high when compared with the theoretical, we see that all meet the conditions described in (2) and (3). They fall in the A_1/A_3 ratio of 1.5 to 2.0 and the distance from hood face to throat is from 0.1 to 0.4 (one 30 deg hood is at 0.5) the face diameter.

There remains for interpretation the values for the larger angles 90 deg, 115 deg, and 136 deg, at the ratios A_1/A_3 of 3, 4 and 5.

The energy loss resulting from contraction of the air stream at the throat of a hood becomes greater with increasing angles, and may become an important proportion of the total loss of large angles. The coefficients found for the large angles are fairly constant for all high ratios, at 0.95, 0.94 and 0.89 for 90 deg, 115 deg and 136 deg, respectively.

This independence of the coefficients relative to area ratios indicates that face velocity head loss is insignificant and that throat loss is, therefore, controlling. Coefficients of contraction of 0.76, 0.73 and 0.66 for 90 deg, 115 deg and 136 deg respectively, give entry coefficients that satisfy the experimental data.

In view of the foregoing considerations we would summarize our interpretation of the authors' data as follows:

1. Very few of the hoods tested are comparable with those for which previous data are published. For the most part they are shorter in taper length.

2. The data are readily correlated with theory by dividing them into three groups: (*a*) Those in which the taper length is greater than 0.6 of the face diameter. (Includes angles only less than 60 deg.) For these the coefficient corresponds to a loss of 0.93 face velocity heads. (*b*) Those in which the taper length is 0.1 to 0.4 the face diameter, and the ratio of face area to duct area A_1A_3 is between, approximately, 1½ to 2. For these the coefficient corresponds to a nearly complete recovery of face-vena-contracta velocity in the duct, giving values of 0.95 to 0.96. (Angles from 20 deg to 115 deg are represented in this group.) (*c*) Those hoods with ratios A_1/A_3 of 3 to 5 and with included angles 90 deg through 136 deg. The coefficients for these hoods appear to be constant, *i.e.*, independent of A_1/A_3 and therefore the loss at the hood throat may be considered as controlling. Coefficients of contraction of 0.76 to 0.66 for these hoods satisfy the data adequately.

B. F. POSTMAN,† New York, N. Y. (WRITTEN): The data presented in this paper will be of practical value to designers of local exhaust systems. For years, designers have utilized their own experience plus a *factor of ignorance* in evaluating estimated losses of the various types of hoods utilized for industrial exhaust systems. Where adequate leeway had been allowed for assumptions or guess, relative to hood losses, the systems usually functioned in a manner which one would normally expect to result from the increase of static pressure allowed for necessary fan performance.

A discussion of the seven effects studied as outlined under *Conclusion* follows:

1. This item is important in that cramped quarters and involved piping are usually present when field tests are to be made of installed systems.

2. This item eliminates one *bogey* since the range of velocities covers the normal field of industrial exhaust application.

3. This item may be the cause for the ineffective results obtainable with many types of hoods installed to date. Structural considerations usually dictate the hood taper allowed for many installations. When these are not a factor, the hood appearance or the personal idiosyncrasies of the sheet metal man, unless constrained by specific dimensions, may be deciding factors determining the shape and size of the hood being installed.

4. This item is more or less a part of item 3.

5. It may be entirely possible that the importance of this item cannot be realized due to the small sized hoods tested. It does become a serious problem when large *fish-tail* hoods are installed on many involved industrial exhaust systems. Perhaps further research will be attempted to cover the larger hood installations.

6. There seems to be a practical limit to the application of flanges to small hoods, particularly portable hoods such as those used for the control of welding fumes. Flanges from 2 to 4 in. in

† Industrial Hygiene Engineer, Employers Mutual Liability Insurance Co. of Wisconsin.

width have been found acceptable by operators who handle small hoods as part of their daily operation procedure. As pointed out in the *Conclusion*, flanges provide an improved air flow pattern in front of the hood. In many instances, their use has cured a *sick* job.

7. There may be wider variations for this item than those indicated if the larger hoods are checked as mentioned under item No. 5 of this discussion. Some very interesting results might be forthcoming if the rule of thumb, 16-1, ratio for large hoods is exploded.

An interpretation of the results presented in this paper, possibly as a corollary to the average curves indicated in Fig. 4, but based on velocity head loss, would materially aid the designer to substitute *factors of research* for *factors of ignorance*.

As stated under *Purpose of Study* it is extremely important that the relationship between the exhaust ventilation rate and the contaminant concentration in the atmosphere be known for different hoods and different operations, if it is to serve as a guide on other similar operations. Research, such as this paper represents, tends to establish a practical approach to the most important phase of industrial exhaust as applied for the protection of the health of industrial workers.

LESLIE SILVERMAN,†† Sc.D., Boston, Mass. (WRITTEN): I believe that Dr. Brandt and Mr. Steffy have presented data which will be of considerable value in the design and checking of industrial exhaust systems. It is interesting to note, from the data, the influence that turbulence at the inlet has on the coefficient of flanged duct ends. In Table 6, where the effect of flanged width is tabulated for different flange widths, it can be noted that the narrower flanges, particularly the round type, number 18F, create higher coefficients for duct ends. I believe this may be explained by the diameter of the vortex created at the mouth of the inlet. The narrower the flange, apparently, the smaller is the radius of the vortex and hence lower inlet restriction and turbulence losses result. I am glad that the authors point out, however, that the entrance loss is not the only consideration in the selection of a hood, since the velocity characteristics in front of the opening are important. In the case of the narrower flange it can be predicted that it will not be as effective in velocity control in front of the opening.

It is of interest to note in Table 2 that the error made in air flow calculation with measurement at one pipe diameter is not serious. A comparison of the square roots of the two suction columns shows an error of 10.7 per cent for the unflanged round duct end, and 2.5 per cent for the flanged round duct end. These values represent the maximum error if only one pipe diameter is used and for field measurements, where length is limited, a correction factor could be applied if desired.

H. H. VALIQUET, Milwaukee, Wis. (WRITTEN): This research on losses through suction hoods has been comprehensive and practical in dealing with sheet metal fabrications reasonably representative of common practices. I saw many of these hoods on a recent visit to the Research Laboratory. My courteous reception by the staff as well as the frank discussion of the test results was appreciated.

There has been great need for this research as I pointed out to the Chicago Section of the *American Industrial Hygiene Association* in a paper entitled, Some Notes on Grinding Ventilation, etc., from which I quote: "The ratio of velocity pressure to total head in the pipe has been 0.5 and has not changed since published about 1910." This produces a coefficient of about 0.71 commonly used for grinding hood ventilation about which I was speaking. "It is not infrequently accepted, without just reason, for many dissimilar orifice arrangements. Tables of air flow are frequently shown in trade literature whether or not pertinent to the apparatus involved."

To ventilating engineers dealing with designs of reasonably correct controls for noxious and toxic substances this research is enlightening and very useful. An example of its value would be in fume hood exhaust design for welding hood No.

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†† Assistant Professor of Industrial Hygiene, Harvard School of Public Health.

18F-2, where a coefficient 0.82 seems to apply to standardized flexible fume hood devices commercially available to which the manufacturer applies an 0.78 energy recovery.

In another field, that of banker exhaust for removing granite dust while finishing monuments, a coefficient of 0.72 is used which agrees with hood No. 18-1 coefficient 0.717. These research data will call attention to hood No. 1-1.2 the coefficient 0.823, of which should be attained in banker exhaust. One of the many facts brought forth in this excellent paper concerns hood No. 17-2 having a coefficient of 0.964, which provides an inexpensive fabricated sheet metal construction having a flow value equal to that of carefully machined ¾ thick orifices used on certain routine tests in high velocity exhaust orifices.

A very significant reference is made to the suction readings being reliable when taken one or more pipe diameters downstream from throat of hood. Where satisfactory performance is determined by use of suction readings, this distance is subject to local restrictions and some judgment.

Users of data presented in this paper will take into account many factors, such as unfavorable turbulences and mechanical obstructions, in adjusting their application of these fundamentals to special controls. There is, however, a large field for immediate application and adaptation of these data to uses in industry and they are invaluable where that delicate balance of control must not interfere with the mechanics of the process being performed.

W. N. WITHERIDGE, Detroit, Mich. (WRITTEN): It is a very great pleasure for me to comment on this history-making research report. As many of you know, I have maintained for some years that the Society's Research Laboratory has been overlooked far too long by engineers in the field of industrial hygiene. In Boston last year I expressed the opinion that industrial hygienists who did not understand the fundamentals of air flow were unwittingly using and perpetuating information that had never been subjected to a really well integrated study using sound methods by ventilating engineers who were competent to question the outworn data of earlier years.

Perhaps the best example of erroneous data that had enjoyed constant repetition in the literature is the implication that hoods designed with long tapers having included angles of 13 to 15 deg are the most efficient from the standpoint of energy loss. It is highly gratifying to find that the very common 45 deg taper, or 90 deg included angle, is at least as efficient as the 7 deg taper, or 14 deg included angle. As pointed out by the authors, this should be quite a relief to those sheet metal fabricators and hood designers who have rarely enjoyed the luxury of long entrance tapers.

It is high time also that the 0.71 entry coefficient be discarded. Fig. 4 in the report certainly indicates that the most likely range of coefficients for conventional industrial ventilating hoods is about 0.80 to 0.95. It also demonstrates that field inspectors who use static suction at the hood throat for estimating the rate of ventilation need more than ever to be advised that exhaust hoods producing the highest air flow rates are those with the lowest entrance losses and consequently the lowest static suction, other factors remaining equal. When a state law can specify a required static suction of five inches of water at the hood throat, without any regard to hood design, it is imperative that engineers fortify themselves with competent data as supplied by this report for the purpose of removing such uneconomical nonsense from the control of uninterested legislators.

I sincerely hope that this study is the beginning of a long and fruitful project in A.S.H.V.E. Research on *industrial process ventilation*. It is not necessary of course that all such work be done at our own Laboratory, but it most certainly must be wisely coordinated by persons experienced in the field through the mechanism of the Society's Technical Advisory Committees and the Committee on Research. The

Laboratory is to be commended for its tenacious pursuit of the general problem that was exposed by the view of many of our members at the special conference on this subject held in Detroit on March 13, 1945. We certainly have ample evidence from the survey conducted by the Special Advisory Committee on Industrial Ventilation that engineers around the country are hungry for sound and reliable design data, and will be ready to support any future work in this field with enthusiasm, and I hope, also with financial aid.

J. N. LIVERMORE, Detroit, Mich.: It would be interesting to know what type of pressure tips were used in making these observations.

A. C. STERN,* New York, N. Y.: The authors are to be congratulated on an excellent research job and a useful presentation of their results. It is hoped that the work can be extended to cover hoods with wheels or tools in position so as to partially obstruct them in the manner most commonly found in practice.

It is presumed that the static pressure taps employed in this study were carefully drilled and finished smooth inside. In the field it is common practice to obtain exhaust hoods static pressure readings from holes made by hammering the point of a punch through the duct wall. This causes internal denting of the duct and internal roughness.

It would be of interest to have the authors discuss the extent that air flow into a hood will differ when the same apparent static pressure is read by a manometer with each of these two kinds of static pressure taps.

AUTHOR'S CLOSURE: The pressure taps used in this study were made by soldering ¼ in. inside diameter brass tubing about 1½ in. long over each hole in the test pipe. The holes were drilled carefully with a 1/32 in. drill, and the inside burr was removed with emery cloth. This procedure is in accordance with that recommended by the Society in THE GUIDE. If the holes in the pipe or duct are made by punching, the results may be fairly accurate or in considerable error depending upon the nature of the burr or rough edge on the inside wall, and upon the depth of the dent produced by the punch.

I am grateful to Mr. Hemeon for his excellent theoretical analysis of the results, which, it is gratifying to note, checks the test results remarkably well. Certainly, his discussion is a most worthwhile contribution to the subject matter of the paper. I regret, however, to state that I am not in complete agreement with one or two of his statements. For example, we did not overlook taper length even though we did not analyze our data on that basis as he did so well. Taper length and face area or face diameter are so inalterably related that you cannot change one without changing the other unless you alter also the rate of taper. It is, therefore, impossible to have an L/d_1 ratio (see Fig. 7) of more than 1 except for small included angles unless the face area to throat area ratio is increased well above 5. The relationship of L/d_1 to the included angle for an area ratio of 5 is given in the following table:

L/d_1	Included Angle in Deg
1	31
1.5	21
2.0	16
2.5	13

Obviously, there is no point in increasing the area ratio beyond 5, and from the results in our paper it becomes questionable whether there is any practical value in

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studying hoods of such small taper rate that L/d_1 can be increased significantly above 1. In this connection, and in response to Mr. Postman's reference to the rule-of-thumb 16-1 ratio, it may be of interest to point out that we obtained for one extreme hood data which were not included in our paper. This hood had a 4 in. diameter throat, a 27 in. face, a 40 deg included angle, and a taper length of 31½ in. These dimensions make $L/d_1 = 1.17$ and the face area to throat area ratio 45. The static suction at one throat diameter differed by only two or three hundredths of an inch of water from that for the similar but much shorter hood listed in Table 7.

The other statement in Mr. Hemeon's discussion with which I cannot agree is that marked 1 in his summary. We specifically tested unflanged round hoods similar to 16 of the 19 given in the table in his paper (our Bibliography 5). Also our hoods were comparable to those referred to in our Bibliography 4. What is more, they represent the type and size of hoods which are in very common use in industry.

There appears to be little need for further comment either on the paper or on the excellent discussions except to thank the reviewers for their kind comments. If we can be guided by the remarks of these reviewers, it is very apparent that there is a great need for data of this kind by industrial hygiene engineers who interest themselves in controlling atmospheric contamination. This is only the beginning of what I hope may be a long series of similar studies which will provide the factual information needed to convert the engineering control of occupational diseases from an art to a science.

1296

No. 1296

CHARACTERISTICS OF UNIT DUST COLLECTORS

By Arthur C. Stern,† Jack Baliff,* Arthur E. Perina,* Robert Crowley,* Benjamin Feiner * and Arthur A. Urbano,* New York, N. Y.

DURING the past four years, 28 makes of unit dust collectors have come to the authors' attention. Their common characteristics are:

1. They combine within the same unit both the means for exhausting the dust laden air and the means for separating the dust therefrom.
2. They are designed for indoor installation immediately adjacent to a dust producing machine.
3. They are small, light and simple enough so that they can be moved and installed by ordinary workmen.
4. They are designed to recirculate the effluent air back into the room in which they are installed.

A wide variety of dust separating means, as shown in Table 1, is employed in the various units, the most common being the replaceable unit filter of the type used in the filter banks of air conditioning and ventilating systems. These filters may be used as either primary or secondary separators. Other units employ cyclones as primary separators and cloth bags or screens as secondary separators. In some units the dust separating means are located on the suction side of the fan; in others on the pressure side; and in still others the fan is located between the primary and secondary separators. A few manufacturers build units of one size only, but the majority build them in two, three or four sizes,.a typical series being units rated at 250, 500 and 1000 cfm, respectively.

Specifications for Ideal Unit Collector

In general, a unit collector should meet the following six requirements:

1. It should maintain not less than its rated air flow capacity in cfm throughout the collection of a reasonably large quantity of dust without requiring servicing, such as cleaning of air filters or emptying of hoppers.

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* Engineer, New York State Department of Labor.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Montreal, Que., Canada, June 1946.

2. Its effluent air should at all times have a dust count no higher than that of the atmosphere outside the factory building.[1]
3. Cleaning of the filters or emptying of the hoppers should be simple and should increase neither room dust concentration nor effluent air dust concentration.
4. It should be quiet in operation.
5. It should be of substantial construction so that it will continue to meet the four previously stated requirements throughout normal use and operation.
6. It should not require: (*a*) Filter rapping or shaking more frequently than once per eight hour day; (*b*) hopper cleaning or filter brushing more often than once a week; (*c*) filter removal for replacement, washing or cleaning, more often than once a month.

STANDARDS FOR TESTING UNIT COLLECTORS

In order to standardize testing of unit collectors against the foregoing criteria, a dust loading of 1.5 grains of dust per cubic foot of air was selected as standard and a test dust was obtained from a foundry tumbling mill exhaust

TABLE 1—MEANS OF DUST SEPARATION EMPLOYED IN COMMERCIAL UNIT DUST COLLECTORS

MEANS	NUMBER OF UNITS
Centrifugal Separator + Replaceable Air Filter	7
Cloth Screens or Bags (alone)	6
Replaceable Air Filter (alone)	5
Liquid Spray or Impingement (alone)	5
Centrifugal Separator + Cloth Screens or Bags	2
Centrifugal Separator (alone)	2
Centrifugal Separator + Liquid Impingement	1
Total	28

system. The size characteristics of this dust are shown in Table 2 and Fig. 1. It will be noted that on a weight basis the mean size of this dust is 140 μ (microns). A microscopic analysis showed this dust to have a mean size of 3 μ on a particle count basis. It is obviously impossible to select a test dust which will represent all possible dusts that a unit collector may be called upon to collect. Since unit collectors are usually sold from stock by retail machinery dealers, machinery jobbers or sales agents, the manufacturer has little control over the type of dust which each unit may be required to collect. Therefore, units tested and approved against a coarse dust might give rise to a nuisance and a potential hazard when used in collecting a fine dust. However, units tested and approved against a fine dust would also do a good collection job on coarse dust. Therefore, a relatively fine dust should be used in testing unit collectors if the tests are to be used as the basis for approving them for widespread indiscriminate use. The dust selected for these tests has the greater part of its bulk in coarse sizes and will therefore not plug a filter as rapidly as if most of it were fine dust. This allows a fair determination of the length of time required before the need for filter shaking, cleaning or replacement. However, it does have a small percentage of quite fine dust which allows its use in determining the effectiveness of units in collecting fine dust.

[1] See Appendix.

TABLE 2—SIZE DISTRIBUTION OF DUST USED FOR TESTING UNIT DUST COLLECTORS

(By weight)

SIZE μ (MICRONS)	PER CENT OF STATED SIZE[a]	PER CENT LESS THAN STATED SIZE
420	7.7	92.3
175	33.6	58.7
150	9.9	48.8
75	22.1	26.7
40	19.0	7.7
30	3.7	4.0
20	1.7	2.3
15	0.8	1.5
10	0.4	1.1
7.5	0.35	0.65
5	0.25	0.40
4	0.17	0.33
1	0.18	0.15
<1	0.15	....

[a] Includes particles from stated size up to next stated size.

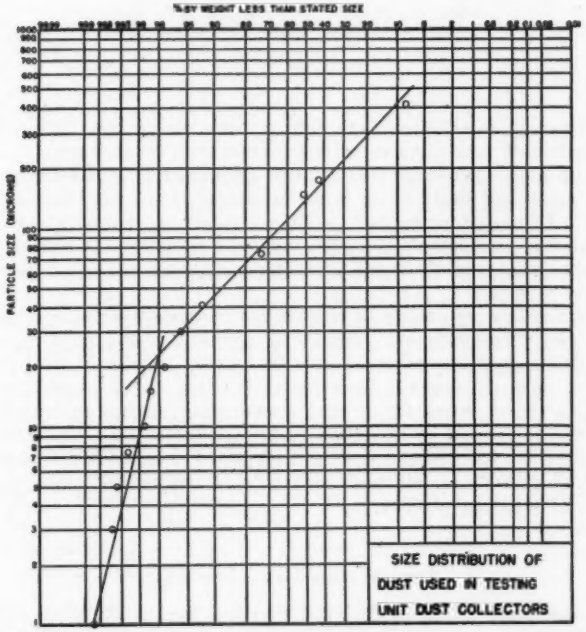

FIG. 1. SIZE DISTRIBUTION OF DUST USED IN TESTING UNIT DUST COLLECTORS

It is assumed that a unit will be under dust load 25 per cent of the working time. There is no factual basis for the choice of 25 per cent as the working time factor. It was based upon a knowledge that dust producing operations such as grinding, buffing, sanding, sawing, container filling, etc., are generally of an intermittent character and actually produce dust during only a percentage of the working time. The 25 per cent assumption is higher than will be encountered in many operations but is, in the authors' opinion, reasonable for rating purposes. Under this assumption, the requirement that a unit should not need filter rapping or shaking more often than once per eight hour day is tested by a two hour continuous test at 1.5 grains per cubic foot loading. Similarly, under this assumption, the requirement that a unit should not need

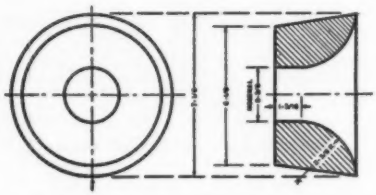

FIG. 2. DESIGN OF POLISHED HARDWOOD FLOW NOZZLE USED IN METERING PLENUM

hopper cleaning more often than once a week means that during a six day week the hopper must have capacity to retain the dust contained in a 12 hour flow at 1.5 grains per cubic foot or 1080 grains per cfm of capacity. This is roughly equivalent to 15 lb or ½ cu ft of dust for every 100 cfm of rated capacity. Filters which cannot be either rapped or cleaned should be under the foregoing assumptions, subjected to a 50 hour test at standard loading. However, for reasons which will become obvious later in this paper, no 50 hour tests were necessary.

That 1.5 grains per cubic foot is a loading commonly encountered in industry may be easily demonstrated. An 18 in. grinding wheel requires exhaust at about 440 cfm and will grind away from 1/10 to 1 cu in. of steel per minute, depending upon the rate at which the work is fed to the wheel. At this rate of exhaust each 1/10 cu in. of steel removed per minute will produce a dust loading in the exhaust of about 0.5 grains per cubic foot. Therefore, the 1.5 grains per cubic foot loading used in these tests is equivalent to grinding at the rate of ⅓ cu in. per minute, which is well within the range of ordinary commercial grinding practice.

TEST METHODS AND EQUIPMENT

Test equipment has been designed to allow the simultaneous measurement of: (1) Rate of air flow in cfm; (2) rate of dust feed in grains per minute; (3) dust count of effluent air, mppcf (million particles per cubic foot); (4)

inlet static pressure of the unit, inches of water; (5) pressure drop across each dust collecting element in the unit, inches of water.

Rate of air flow was measured by means of rounded approach flow nozzles, Fig. 2. Seven such nozzles, made of hard wood, turned to accurate profile, shellacked and polished, were inserted in the front, top and side walls of a large plenum chamber, Figs. 3 and 4. This plenum chamber is made of shellacked plywood glued to a heavy wooden frame to prevent leakage at the joints. The joint between the tapered exterior of each flow nozzle and its hole in the wall of the plenum was sealed with a plastic sealing compound. The number of flow nozzles in use at any one time is controlled by inserting rubber stoppers into the orifices of nozzles which are to be closed. By this means, it was possible to vary, from one to seven, the number of nozzles through which air flows to the plenum. Test practice has been to close side wall, rather than front wall nozzles, whenever three or fewer nozzles are to be closed.

Although these nozzles are nominally of 2⅜ in. throat diameter, each has been carefully calipered several times and the actual, rather than nominal, diameter used in calculating air flow. The pressure in the plenum was measured by a 2 in. inclined tube draft gage. No temperature or barometer corrections were made, insofar as all flow computations were made using the formula:

$$Q = CA\ 4005\ \sqrt{h}$$

Q = cfm at 68 F—30 inches mercury
C = nozzle coefficient—taken as 0.98
A = area of open nozzles in square feet
h = plenum pressure—inches water

One wall of the plenum chamber is equipped with outlet holes of several diameters [2] so as to allow connection to different sized unit inlet pipes. Those not in use are sealed. Those in use are connected by a flanged coupling to a blast gate, which in turn is connected by a length of straight duct to the inlet of the unit collector, Fig. 5, under test. The length used, in most cases, was about five feet in order that the blast gate might not adversely influence the measurement of unit inlet static pressure. This latter measurement generally has been made by placing a pressure tap in the straight length of duct near its junction with the collector. In a few cases, where an inlet plenum was a part of the collector design, the inlet static pressure tap was taken from this plenum. Inlet static pressure was measured by a U tube water gage.

Location of pressure taps to measure pressure drop across the various dust collecting elements varied according to unit construction. Centrifugal separator pressure drops were measured by a U tube water gage. Air filter, cloth screen and cloth bag pressure drops were measured by an inclined tube draft gage as long as the pressure drop remained within gage limits; beyond that a U tube water gage was used.

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[2] In future designs, it is recommended that low loss rounded approach nozzles be used as outlets rather than the sharp edged orifices used in the author's tests. This will allow lower metering loss than was possible with the plenum design.

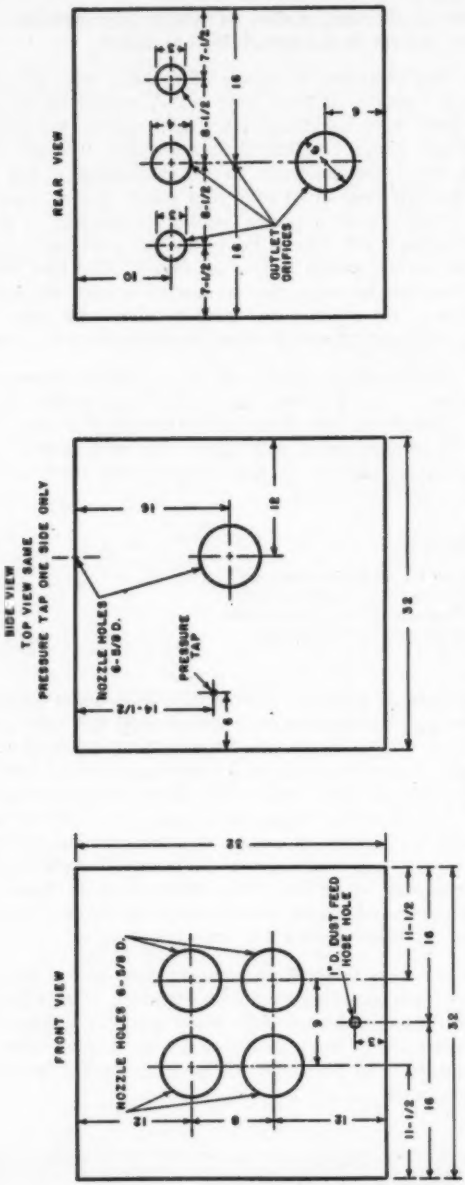

FIG. 3. DIMENSIONS OF PLENUM CHAMBER FOR METERING AIR FLOW DURING TESTS OF UNIT COLLECTORS

DUST FEEDING PROCEDURE

Dust feed to the unit under test was by means of a No. 2 Hancock ejector (Fig. 6) operated by compressed air at 8-10 psi. A member of the test crew manually fed dust into a funnel connected to the suction inlet of the ejector at a constant measured rate using various sizes of scoops, the level capacity of which had been previously determined. Feed rates were varied by varying the scoop size and maintaining wherever possible the rate of four scoops per minute. Experience shows that this procedure maintains a uniform (and easily varied) rate of feed and is preferable to automatic feed where time required for feeding is not too great. It is necessary to prevent the ejector nozzle

FIG. 4. PHOTO OF PLENUM CHAMBER FOR METERING AIR FLOW

from clogging in order to maintain dust feed. This requires exclusion from the feed of extraneous matter such as splinters, paper, scale, etc., and this is easily accomplished by placing a coarse mesh screen over the funnel into which dust is fed.

The discharge from the ejector carrying the test dust in suspension was piped through rubber tubing through the front wall of the plenum chamber and terminated in front of the center of the opening to the duct from the plenum to the unit under test.[3] In this manner the suspended test dust was blown into the duct leading to the unit being tested. In order to avoid the complication of having to measure the air used to transport dust, the test procedure was to shut off the air flow through the aforementioned rubber

[3] In future designs, it is recommended that the ejector discharge be injected into the duct from the plenum to the unit under test at a point after the blast gate by means of a branch connection to that duct. This will lower the metering loss by removing the obstruction created by the presence of the dust tube in front of the plenum outlet inside the plenum chamber.

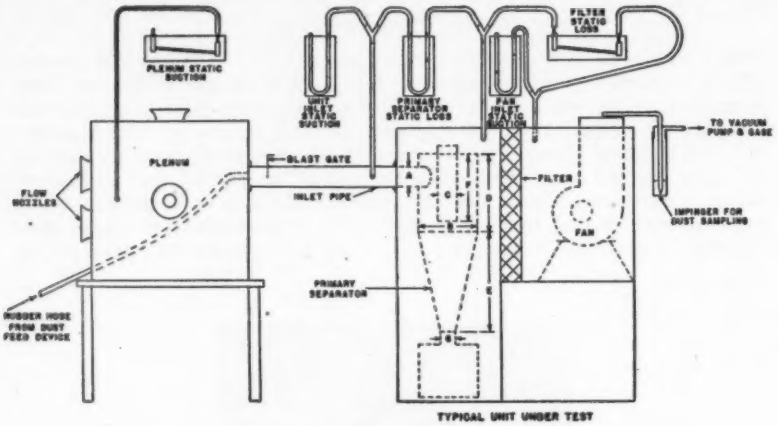

FIG. 5. SET UP FOR TESTING UNIT DUST COLLECTORS

tubing whenever it was desired to measure air flow into the unit under test. Because of this procedure, dusting tests actually consisted of a series of runs with the compressed air injection in operation while dusting, and turned off while metering. The unit, of course, ran continuously during the test regardless of whether or not compressed air feed was operating. Since all test results reported herein are based upon data obtained during metering runs with compressed air turned off, they are the same as would have been obtained by any other method of supplying dust. The quantity of compressed air supplied each unit while dusting was negligible in comparison with the quantity of air handled by the unit in ordinary operation and had no adverse effect upon the results of these tests.

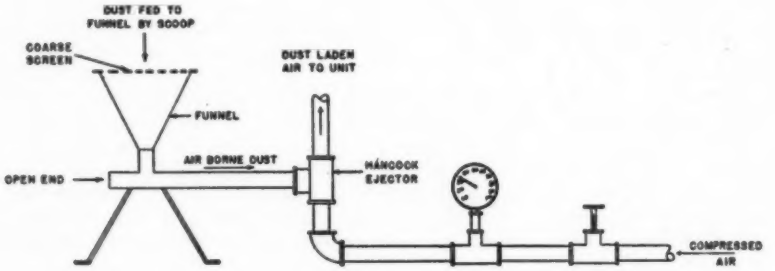

FIG. 6. METHOD OF FEEDING DUST TO UNIT UNDER TEST

TEST PROCEDURE

Test procedure was generally to measure all static pressures and air flow through the unit first with the blast gate wide open and then with it closed by successive steps from fully open to fully closed. Several *clean air* check runs of the unit were made and plotted before *dusting* any of the units tested. Following this, all units containing filters or cloth surfaces designed for rapping or shaking, rather than replacement, were subjected to several unmetered *dirtying* runs, each followed by a rapping or shaking of the filters or bags, in order to change the filtering element from the clean to the *shaken* condition. After bringing the unit to the *shaken* condition, the final series of *dusting* runs were undertaken. As has already been noted, a dusting run consisted of alternately dusting and measuring air flow and pressures. As a rule units were dusted ten minutes after which the compressed air was turned off; the taking of pressure readings generally required from one to five minutes of time, depending upon whether readings were taken only with blast gate fully open, or with various degrees of throttling. Following the taking of pressure readings, the compressed air was turned on again and dusting resumed for another cycle. These cycles were continued for two hours at a dust feed rate of 1.5 grains per cubic foot [4] unless an earlier test end point was indicated. This was desirable when (*a*) the collector hopper became filled before the end of the two hour period; (in the case of centrifugal collector hoppers, the hopper was considered filled when the level rose to the height at which reentrainment of dust from the hopper occurred); or (*b*) when dust visibly escaped from the unit outlet (in some units tested, dust escaped visibly from the very start of the test, and runs were, therefore, stopped as soon as it became evident that further operation would not improve the situation).

When counts were made, the effluent air was sampled by means of a Greenburg-Smith impinger, at the rate of 1 cfm for 10 min. Counts were made by the standard low power-light field technique. Counts were not taken when there was visible effluent dust from a unit since this alone was taken as indicating excessive dust count of the effluent air. In several collectors where this occurred, the effluent dust came largely from around filters rather than through them. However, since collectors were being tested in the *as manufactured* condition, very little effort was made to improve the fit of filters or make other changes in the unit. The only modifications made in the collectors tested consisted of (*a*) adjusting a fan rotor if it rubbed the casing when received, (*b*) stiffening the casing back of a fan rotor to prevent its collapse inward against the rotor when the unit was placed under a static no delivery test, and (*c*) adding a gasket to a filter frame of a unit to allow its test to be run to completion.

PLOTTING TEST DATA

The test data from each test run were routinely plotted as follows:

1. Pressure drop across centrifugal collector in inches water against air flow in cfm, Fig. 7.

[4] Accelerated loading was used for the last half of some tests by feeding dust at rates two and four times 1.5 grains per cubic foot for a half or quarter hour respectively so that the total weight of dust fed remained the same as if the 1.5 grain rate had been maintained for the hour.

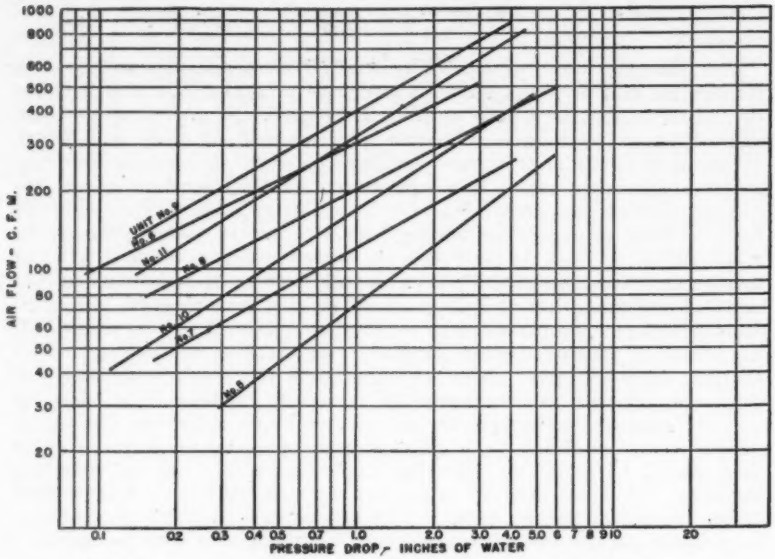

FIG. 7. AIR FLOW *vs.* CYCLONE PRESSURE DROP FOR VARIOUS UNITS

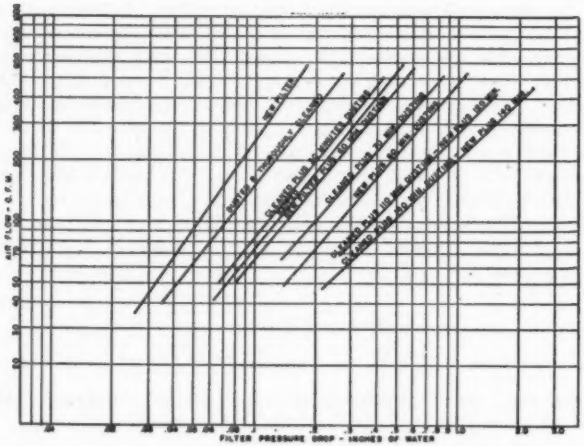

FIG. 8. TYPICAL FAMILY OF CURVES OF FILTER PRESSURE DROP *vs.* AIR FLOW FOR UNIT NO. 10

2. Pressure drop across filter or cloth in inches water against air flow in cfm, Fig. 8.

3. Inlet static pressure in inches water against air flow in cfm, Fig. 9.

In all instances, the pressure drop *vs.* air flow plots yielded straight lines on double logarithmic coordinates. In the case of centrifugal collectors, one such line represents both clean and dirty operation, Fig. 7, whereas for filter or cloth there is a different straight line for each state of dirtyness, Fig. 8, *i.e.*, the line shifts as the unit collects dust. Use was made of these graphs to determine the maximum air flow obtainable with a unit collector disconnected from the metering plenum, inlet duct and all other external resistance to air

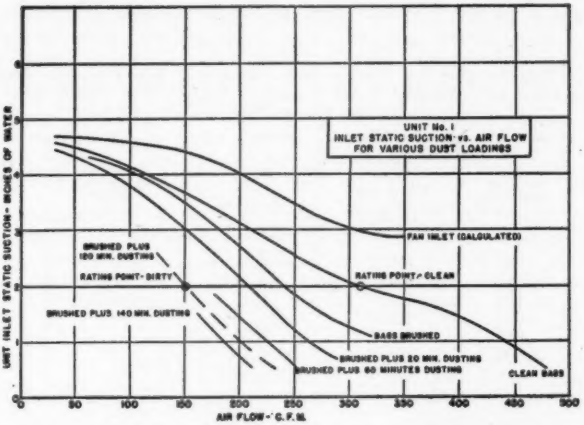

FIG. 9. INLET STATIC SUCTION *vs.* AIR FLOW FOR VARIOUS DUST LOADINGS FOR UNIT NO. 1

flow. When the unit was operated in this condition, the pressure drop across its centrifugal collector or filter elements was measured and the air flow corresponding to that drop obtained by extrapolation of the *pressure drop-air flow line* previously plotted for the unit.

DETERMINATION OF INLET STATIC PRESSURE

In the test set up described, the pressure drop ahead of the inlet static pressure tap consisted of four parts: (1) the drop through the metering flow nozzles; (2) the drop at the inlet of the duct from the plenum to the unit; (3) the friction loss in that duct; and (4) the throttling loss past the blast gate in that duct. This latter drop disappears when the blast gate is full open, as is the case when measuring maximum metered flow. The minimum drop across the metering flow nozzles occurs when all seven nozzles are open and in use. Dust drop and drop from plenum to duct may both be estimated.

Table 3 shows for each size of inlet duct the cfm required at 4000 fpm; the pressure drops in metering this cfm using all seven orifices; in passing it through straight duct to the unit with blast gate open; and in entrance to this straight duct. The table also shows the sum of these three pressure drops plus 1 in. velocity head, equivalent to 4000 fpm. This sum, therefore, represents the theoretical minimum inlet static pressure measurable by our test setup at 4000 fpm unit inlet velocity. This analysis has been based on 4000 fpm velocity because the New York State Dust Collecting Code (Code Bulletin No. 12) requires a minimum of 2 in. static suction in the branches of dust collecting systems. This presumes a branch velocity of 4000 fpm and a hood of average construction having an entrance loss equal to one velocity head in the branch.

It will be noted from Table 3 that the test setup, therefore, theoretically allows units to be tested at very close to the code requirement of 2 in. static

TABLE 3—COMPUTATION OF THEORETICAL MINIMUM INLET STATIC PRESSURE OBTAINABLE AT 4000 FPM INLET VELOCITY USING TEST SETUP EMPLOYED IN THESE TESTS

DUCT DIAMETER IN.	CFM AT 4000 FPM	PLENUM DROP WITH ALL 7 NOZZLES IN. H_2O	DUCT DROP IN. H_2O	DROP ON INLET TO DUCT IN. H_2O	THEORETICAL MINIMUM INLET STATIC PRESSURE* IN. H_2O
3	196	0.059	0.42	0.47	1.95
3½	267	0.110	0.33	0.47	1.91
4	349	0.187	0.28	0.47	1.94
4½	440	0.298	0.24	0.47	2.01
5	544	0.455	0.21	0.47	2.14
5½	660	0.670	0.18	0.47	2.32
6	786	0.950	0.16	0.47	2.58

* Includes 1 inch velocity pressure.

suction and, by throttling, allows testing at all higher inlet static pressures up to the static no delivery point of the unit. In actual test setups of units having the various sized inlet ducts in Table 3, maximum (unthrottled) inlet duct velocities both higher and lower than 4000 fpm were found. In most cases where the velocity was lower than 4000 fpm, the minimum measurable inlet static suction was lower than that shown in Table 3. Conversely, where the maximum velocity exceeded 4000 fpm or where pressure losses additional to those considered in Table 3 were met (as, for instance, where elbows had to be used to connect the plenum to the unit under test), the minimum measurable inlet static suction was higher than that shown in the table. Whenever the minimum measurable inlet static suction was in excess of 2 in., it was necessary to determine the air flow at 2 in. inlet static pressure graphically, by extrapolating the inlet static pressure-cfm curve of the unit in question until it crossed the 2 in. pressure line. The corresponding value of cfm could then be obtained from the curve and used as the rating point of the unit, see Fig. 9. Fortunately, this procedure was required in only a few cases because most of the test runs included a reading at or below 2 in. inlet static suction.

BASIS FOR RATING UNIT COLLECTORS

Rated cfm of a unit collector as based upon these tests is, therefore, the flow at 2 in. inlet static after the unit has accumulated dust for two hours as previously noted. This flow may be one that was actually measured, or one extrapolated to 2 in. inlet static in those cases where the test data did not drop that low in inlet static. The cfm rating could not be established on this basis for any unit in which the hopper became filled or from which dust visibly escaped before the end of the 2 hour dusting test. However, units in which this occurred were not acceptable, *per se*, and a cfm rating would, therefore, be meaningless and misleading if given. It should be noted that free inlet air flow has no useful significance, since it can never be realized in practice. It should also be obvious that the free inlet flow decreases as the filter or cloth elements become more dirty. Despite these facts, the practice has arisen of rating commercial unit collectors on their free inlet flow in the clean condition. One of the principal objectives of the tests herein reported was to determine how far the usable capacity (*i.e.*, the 2 in. inlet static dirty rating) of these units deviated from their published free flow clean rating.

CHARACTERISTICS OF UNIT COLLECTORS BASED ON MANUFACTURER'S RATING

To date 18 unit collectors of 13 different makes have been tested. Although the test program is continuing, the makes tested represent over 90 per cent of the unit collectors sold in New York. Physical characteristics of the types tested are shown in Table 4. For purposes of analysis certain derived data as given in Table 5 are easier to interpret than the data of Table 4.

MOTOR HORSEPOWER PER 1000 CFM

From Table 5 it will be seen that among the units tested, motor hp per 1000 cfm manufacturer's rated capacity varied from 0.35 to 3.3.

A unit rated at 2 in. inlet static pressure will require at least an additional 2 in. static pressure to drive the air through its dust collection device when the latter is in the dirty condition. It is, therefore, apparent that the fan static pressure must in almost all cases be at least 4 in. At 4 in. static pressure, the required hp per 1000 cfm is equal to or greater than 1.3. Despite this fact, it will be seen that 6 of the 18 units tested had motor hp per 1000 cfm of under 1.3 and are obviously underpowered for the manufacturer's claimed rating even without testing the unit to prove the fact.

FILTER VELOCITY

Similarly it will be seen that filter velocities based on manufacturer's rated capacity range from 3.8 to 180 fpm. In the case of those units having filter velocities of 3.8, 5.0 and 8.3, respectively, the velocity is based upon the gross cloth area of the unit. However, by observing the bags of these units after extensive accumulation of dust, it was seen that only about one-third of the cloth area became dirty, the remaining cloth surface being in such close contact with adjacent cloth surface that it was not actually available as filter surface.

TABLE 4—PHYSICAL CHARACTERISTICS OF UNIT COLLECTORS TESTED

UNIT NO.	MFR'S. RATED CAPACITY CFM	FILTER AREA SQ FT	HOPPER CAPACITY CU FT	MOTOR HP	FAN RPM	FAN TYPE[a]	INLETS (AS TESTED) IN.	FILTER MEDIA
1	500	60	0.44	½	3450	FC	2–3	Cloth bags (steel wool filled)
2	500	60	0.44	½	3450	FC	2–3	Cloth bags (steel wool filled)
3	600	120	0.45	1	3450	FC	1–6	Cloth bags (steel wool filled)
4	1000	45	3.0	1½	3450	FC	2–3	Cloth bags
5	325	4	0.18	¼	3400	FC	1–3	Hair glass mat
6	450	120	2.3	1	3450	FC	1–4	Cloth bags (steel wool filled)
7	500	3½	0.55	¾	3450	FC	2–3	Paper pad filter
8	550	11	1.0	¾	3600	FC	1–4	Cloth screen
9	880	16½	3.8	2	3450	PW	1–6	Steel wool filter
10	950	7¼	0.23	⅓	3450	FC	1–5	Glass wool filter
11	1250	13	0.23	1½	3450	FC	1–6	Glass wool filter
12	270	28.5	2.75	½	3450	[b]	1–3½	Wool felt screen
13	450	2.5	2.3	1½	4300	[b]	1–4	Metal oiled filter
14	400	0	0.27	½	3450	FC	1–2½	None
15	700	0	[d]	1	3475	PW	[c]	None
16	1000	0	[d]	2	2200	PW	1–8	None
17	350	60	1.37	½	3450	PW	1–4	Cloth bags
18	985	94	1.13	2	3450	PW	1–6	Cloth bags

[a] FC—Forward Curve; PW—Paddle Wheel.
[b] Dynamic Precipitator type combined fan and dust separator. Fan is of PW type.
[c] Directly connected to grinding wheel hood.
[d] Sludge tank.

TABLE 5—DERIVED DATA ON UNIT COLLECTORS TESTED BASED ON MANUFACTURER'S RATED CAPACITY

UNIT NO.	MOTOR HP/1000 CFM	FILTER VELOCITY FT/MIN	HOPPER CAPACITY PER 100 CFM (CU FT)	INLET DUCT VELOCITY FT/MIN
1	1	8.3[a]	0.09	5080
2	1	8.3[a]	0.09	5080
3	1.67	5.0[a]	0.08	3060
4	1.5	22	0.30	10200
5	0.77	81	0.06	6600
6	2.22	3.8[a]	0.51	5160
7	1.5	143	0.11	5080
8	1.36	50	0.18	6300
9	2.28	53	0.43	4480
10	0.35	131	0.03	6950
11	1.2	96	0.02	6360
12	1.85	9.5	1.0	4050
13	3.33	180	0.51	5150
14	0.833	....	0.07	11700
15	1.43	....	....	....
16	2.00	....	....	2860
17	1.43	5.8	0.39	4000
18	2.04	10.5	0.11	5000

[a] Actual velocity approximately 3 times higher because only ⅓ of filter area is effective area.

The actual velocities in these units are therefore about three times that shown in Table 5 and are about 11.4, 15.0 and 25.0 fpm, respectively. Conventional commercial cloth screen and bag dust arresters for permanent installation are designed to operate at from 2 to 4 fpm filter velocity. Even though, in the interest of compactness, somewhat higher velocities may be desirable, it is doubtful if good practice can tolerate such higher velocities except in those cases where either a precleaner of exceptionally high efficiency is used ahead of the filter, or the filter surface is frequently cleaned automatically.

HOPPER CAPACITY PER 100 CFM

Table 5 also shows that hopper capacity per 100 cfm manufacturer's rated capacity varied from 0.02 to 1.0 cu ft. In three units hopper capacity per 100 cfm was over 0.50 and therefore conform to the standard previously

TABLE 6—DIMENSIONS OF CYCLONE SEPARATORS IN UNIT COLLECTORS TESTED (INCHES)

UNIT NO.	NUMBER OF CYCLONES	CYCLONE INLET [c] A	CYCLONE DIAMETER B	INNER CYLINDER DIAMETER C	HEIGHT OF OUTER CYLINDER D	LENGTH OF CONE E	LENGTH OF INNER CYLINDER F	DUST OUTLET DIAMETER G
5	1	3	11¾	4½	5½	17½	10½	2
6	1	7½x2¾	12	6	16	22¼	14	5⅝
7[a]	2	3	8	4½	5¼	11⅝	-	4
8[b]	1	4	16	9	14	0	9¾	0
9	1	6	19	10	7½	26	12	6
10	1	6x3	9	5⅛	11¾	17¾	8	2½
11	1	8x4¼	13¼	7 1/16	18	18	12	4

[a] Two cyclones of same dimensions in parallel.
[b] Cyclone has no bottom cone.
[c] Dimensions as shown on Fig. 5.

noted. The three units with capacity per 100 cfm of 0.30, 0.39 and 0.43 cu ft are not excessively low. The other ten seem to have much less than the desired hopper capacity. It was noted that in all but one of the cloth bag and screen type units, the dust hopper was located in such a position that the incoming air swept across the hopper at relatively high velocity. This tended to prevent utilization of the full hopper capacity by re-entraining some of the finer dust from the hopper after a critical level had been reached.

CHARACTERISTICS OF CYCLONE SEPARATORS TESTED

Seven of the units tested employed a cyclone separator as the primary means of dust collection. The dimensions of these cyclones listed in Table 6 are those shown on the typical cyclone in Fig. 5. The pressure drop versus cfm characteristics of each of these cyclones is shown graphically in Fig. 7. The numbers used to identify the lines on Fig. 7 are the unit numbers listed in the left hand column of Table 6. No correlation has yet been attempted between cyclone dimensions and cyclone pressure drops. However, it is hoped that as more data of this type accumulates as a result of these tests, such correlation may become possible.

TABLE 7—UNIT OUTLET DUST COUNTS[a]

UNIT NO.	COUNT MPPCF	FEED RATE GRAINS/CU FT	UNIT PREVIOUSLY DUSTED AT 1.5 GR/CU FT RATE (EQUIV) MIN	NOTES
1	29.8	0.375	0	Bags rapped
1	29.8	0.75	5	Dusting started after bags rapped
1	67.2	1.5	10	Dusting continued
1	146	3.0	140	Bags loaded with dust
8	39.4	1.5	20	Filter cloth as received (previously dirtied)
8	97.5	1.5	20	Filtercloth after brushing
10	91.2	1.5	10	Filter clean at start
10	57.6	1.5	50	Dusting continued
10	37.4	1.5	80	Dusting continued
10	24.6	1.5	110	Dusting continued
10	19.2	1.5	130	Dusting continued
10	29.4	1.5	10	Filter rapped at start
10	10.4	1.5	50	Dusting continued
10	24.0	1.5	100	Dusting continued
10	13.2	1.5	110	Dusting continued
11	18.8	1.5	220	Filter clean at start—dusted
11	34.6	1.5	350	Dusting continued
11	30.0	3.0	40	Filter rapped at start—dusted
11	17.4	3.0	280	Dusting continued
12	5.0	3.0	100	Additional gasketing placed around filter frame to suppress leaks
15	324	1.5	0	Water level too low—detergent added to water
15	16.0	1.5	0	Water level correct—detergent added to water
15	15.0	1.5	0	Water level correct—detergent added to water
16	24.0	1.5	0	Operating at 57 per cent rated capacity plain water
16	32.0	1.5	0	Operating at 57 per cent rated capacity plain water
16	35.0	1.5	0	Detergent added to water
17	5.6	1.5	10	Bags shaken at start
17	3.8	1.5	40	Dusting continued
17	3.6	1.5	120	Dusting continued
18	3.8	1.5	0	Bags shaken at start
18	1.2	1.5	60	Dusting continued
18	3.0	1.5	120	Dusting continued

[a] Units 5, 7, 9, 13 and 14 had excessive dust in effluent by visual observation.
Units 2, 3 and 6 use same filter media as unit 1: No visible dust from unit 4.

DUST COUNTS IN EFFLUENT AIR FROM COLLECTORS

It was known before these tests were run and confirmed by them that when dust collector effluents are visibly dusty, the effluent air dust count is in excess of 100 mppcf.[5] Thus the effluent of unit 15 which was visibly dusty when the water level was set too low had an effluent dust count of 324 mppcf (Table 7). It was therefore deemed unnecessary to make confirmatory dust counts when collector effluents were visibly dusty. This condition was observed in units 5, 7, 9, 13 and 14 as well as in unit 15 when the water level was too low.

Units 2, 3, and 6 had filter media identical with unit 1 and their effluents were therefore not sampled. The effluent of unit 4 was visibly clean but no

TABLE 8—CHARACTERISTICS OF UNIT COLLECTORS TESTED BASED ON CFM AT 2 IN. UNIT INLET STATIC PRESSURE WITH UNIT CLEAN

UNIT NO.	CFM	PER CENT OF MFR'S RATED CAPACITY	INLET DUCT VELOCITY FT/MIN.	HOPPER CAPACITY CU FT PER 100 CFM	FILTER VELOCITY FT/MIN	MOTOR HP PER 1000 CFM
1	315	63	3200	0.14	5.2[c]	1.59
2	580	116	5900	0.08	9.7[c]	.86
3	850	141	4320	0.05	7.1[c]	1.18
4[a]	700	70	7120	0.43	15.5	2.14
5	140	43	2850	0.13	35.0	1.78
6	430	95	4960	0.54	36.0[c]	2.32
7	415	83	4220	0.13	120.0	1.80
8	315	57	3600	0.32	38.6	2.38
9[a]	1000	113	5100	0.38	60.6	2.00
10	225	24	1650	0.10	31.0	1.48
11	620	50	3160	0.04	47.6	2.42
12	305	113	4560	0.90	10.7	1.64
13	420	94	4820	0.55	168.0	3.57
14[b]	160	40	4700	0.17	....	2.08
15[b]	([d])	...	....	....	....	....
16[b]	400	40	1150	....	....	5.00
17	400	114	4600	0.34	6.7	1.25
18[a]	1800	180	9000	0.06	19.2	1.11

[a] Approx.—Flow measurement beyond range of test set up at 2 in. static pressure.
[b] Clean and dirty rating the same because unit contains no filter to increase resistance.
[c] Actual velocity is approximately 3 times higher because only about ⅓ of filter area is effective area.
[d] Air flow varies with water level: At high level flow is low and dust collection high; at low level, vice versa.

counts were made. As originally tested, unit 12 leaked dust around the filter. However, to determine the efficiency of the filter media, the leaks around the filter were stopped by gasketing and the dust count was then obtained and recorded.

No consistent trend of counts with continuing dusting was observed which is applicable to all units. Units 1 and 8 are of the cloth type. In the former, counts were higher after the bag had a dust layer than before. This is probably due to increased leakage around the filters at high pressure differentials across them. Units 10 and 11 are of the glass wool filter type. In two

[5] Mppcf = million particles per cubic foot.

test runs on unit 10 and one on unit 11, dust counts decreased as the filter became loaded. In one run on unit 11, a reverse trend was noted which may be accounted for by the fact that this occurred toward the end of a run of long duration. It is possible that the dust holding capacity of the filter had been exceeded so that continued dusting allowed greater escape of dust, whereas up to this point the reverse was true. In the case of unit 15, it was observed

TABLE 9—CHARACTERISTICS OF UNIT COLLECTORS TESTED

Based on cfm at 2 in. unit inlet static pressure with unit dirtied for equivalent of 2 hours at 1.5 grains per cu ft loading of dust having mean size of 140 microns on weight basis and of 3 microns on particle count basis.

UNIT NO.	CFM	PER CENT OF MFR'S RATED CAPACITY	INLET DUCT VELOCITY FT/MIN	HOPPER CAPACITY CU FT PER 100 CFM	FILTER VELOCITY FT/MIN	MOTOR HP PER 1000 CFM	EFFLUENT DUSTINESS[b]	PROPER INLET DIAM. INCHES
1	150	30	1530	0.29	2.5[d]	3.34	B	1–2½[g]
2	240	48	2440	0.18	4.0[d]	2.08	A	1–3
3	565	94	2880	0.08	4.7[d]	1.77	A	1–5
4[a]	600	60	6100	0.50	13.3	2.50	A	2–3½
5	[b]	[b]	[b]	[b]	[b]	[b]	D	1–2[g]
6[f]	430	95	4960	0.54	3.6[d]	2.32	A	1–4½
7	[b]	[b]	[b]	[b]	[b]	[b]	C	1–4
8	200	36	2300	0.50	18.2	3.75	C	1–3
9	[b]	[b]	[b]	[b]	[b]	[b]	D	1–6
10	125	13	915	0.18	17.2	2.66	C	1–2[g]
11	505	40	2570	0.05	38.8	2.97	C	1–4½
12	175	65	2620	1.57	6.15	2.86	B	1–2½[g]
13	[b]	[b]	[b]	[b]	[b]	[b]	D	1–4
14[e]	160	40	4700	0.17	....	2.08	C	1–2½[g]
15[e]	[e]	..	....	....	....	....	[e]	....
16[e]	400	40	1150	....	....	5.00	C	1–4
17	240	69	2760	0.57	4.0	2.1	A	1–3
18	660	67	3400	0.17	7.0	3.0	A	1–5½

[a] Approx.—Flow measurement beyond range of test set up at 2 in. static pressure.
[b] Test not carried to completion because exhaust remained visibly dusty during first 20 min of dusting.
[c] Clean and dirty rating the same because unit contains no filter to increase resistance.
[d] Actual velocity is approximately 3 times higher because only about ⅓ of filter area is effective area.
[e] Air flow varies with water level: At high level flow is low and dust collection high; at low level, vice versa.
[f] Test terminated after 80 min dusting with very little decrease in flow.
[g] Lower than is allowable in New York State.
[b] Effluent dustiness rated by three observers on following basis: A—No dust through filter and no leakage around filter; B—No dust through filter but leakage around filter; C—Dust blows through filter but no leakage around filter; D—Dust blows through filter and leaks around filter.

that when the water level was lowered, the visible dust concentration of the effluent increased, and vice versa.

It is noteworthy that only 3 units had dust counts below 5 mppcf. Since one would expect ambient room counts, without recirculation from a dust collector, to be well under 3 mppcf in any room in which a unit collector is installed, it appears that only one of the units tested meets the suggested standard[6] of a maximum dust concentration in the unit effluent of 3 mppcf in excess of ambient dust count.

[6] See Appendix.

TEST RESULTS—CLEAN RATING

The clean ratings (cfm at 2 in. unit inlet static suction before dirtying filter) of the 18 units tested are shown in Table 8, together with some data derived therefrom. On this rating basis, only units 2, 3, 9, 12, 17 and 18 have a capacity equal to or in excess of rated capacity; units 1, 5, 10, 11 and 16 have too low an inlet velocity, and units 2, 3 and 18 too low a motor hp per 1000 cfm. Filter velocities are uniformly too high and, with the exception of units 6, 12 and 13, hopper capacity uniformly too low.

TEST RESULTS—DIRTY RATING

The dirty ratings (cfm at 2 in. unit inlet static suction after dirtying filter) of the 18 units tested are shown in Table 9 together with some data derived therefrom. In the authors' opinion the ratings shown in this table are those upon which use in the field should be based. Because these ratings are in all instances lower than manufacturer's rated capacity and since inlet duct velocity in over half the units is below acceptable limits, it is obvious that these units are as a class not only overrated but also are provided with excessively large inlet duct area. Included in Table 9, therefore, is a column headed *Proper Inlet Diameter Inches,* which shows the inlet diameter necessary to maintain at least 4000 fpm inlet velocity at the dirty rating listed. These are the inlet diameters which, in the authors' opinion, should be provided on these units rather than those actually provided. In five of these units indicated by note g in Table 9, the proper inlet diameter is either 2 or 2½ in. However, none of the New York State dust control codes lists operations where branches smaller than three inches may be employed. *These* particular units are, *therefore,* undersize for all dust control operations.

Table 9 also includes a column headed *Effluent Dustiness* and a footnote showing the rating scale for dustiness. This rating is difficult to correlate with the dust counts of Table 7 since it is qualitative rather than quantitative. With regard to the three wet units without filters which were tested, the rating *C* given means that visible dust blew through the unit.

By way of summary, a bar chart, Fig. 10, has been prepared showing how each unit tested compares with the various arbitrary standards suggested. The preferable units are those which most nearly meet or exceed the most of these standards.

Some performance factors, such as ease and cleanliness of cleaning filters and emptying hoppers, the rigidity of construction and the noisiness of operation had to be determined objectively. If the importance of quietness of operation had been realized early in these tests, noise level on each unit tested could have been measured. Unfortunately, this realization came slowly as more and more noisy units were tested. Specifications for unit collectors should include a limiting noise level.

CONCLUSIONS

These tests have shown a real need for a standardization of rating and sizing procedure because they have revealed a wide divergence between manufacturer's rated capacity (free flow into a clean unit) and actual usable capacity

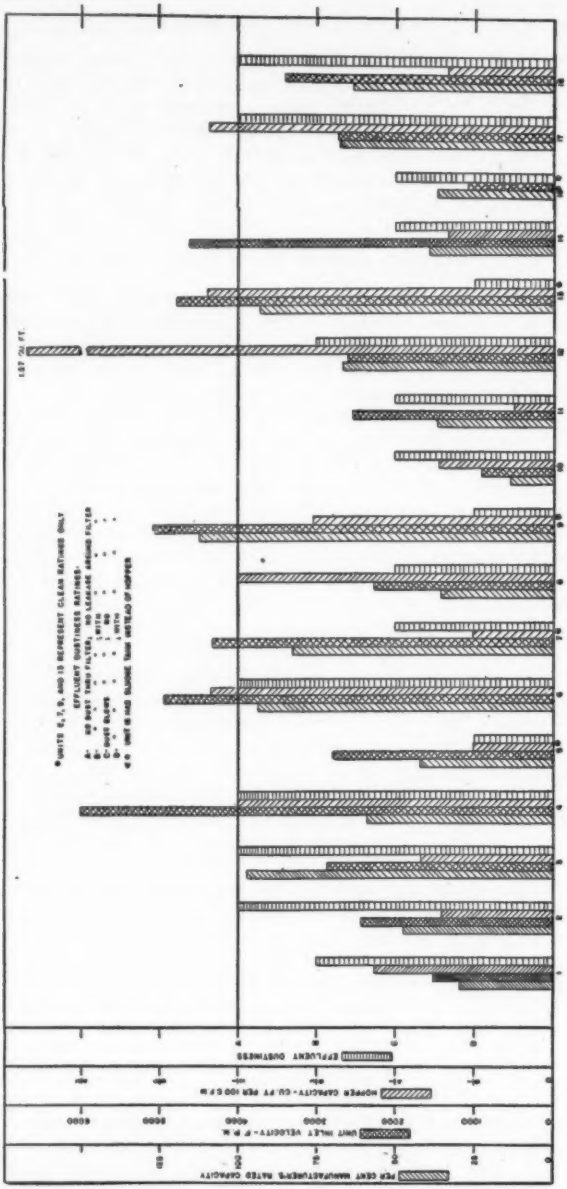

FIG. 10. COMPARATIVE PERFORMANCE OF UNIT COLLECTORS

(flow through hoods and ducts into a dusted unit). For the protection of the purchaser, it would seem desirable that once a standardized rating procedure is adopted, either by the manufacturers themselves or by the various state regulatory bodies, units be given nameplates rated from performance data based on such rating procedure.

Recirculation of air from a dust collector back into the workroom can be successfully accomplished only under exceptionally well controlled conditions. These tests have confirmed this conclusion, since only six of the eighteen units tested had a visible clean effluent and only one had an acceptable effluent dust count. In the remainder, appreciable dust either blew through the filter or leaked out around the filter. None of these latter units are acceptable for recirculation regardless of the adequacy of their air flow rating and inlet sizing. They might be useful, however, if provided with a discharge duct connection and adequate additional fan capacity so that the effluent could be discharged out of doors instead of back into the workroom.

SUMMARY

1. A set of proposed standards specifications for unit dust collectors has been prepared.
2. A total of 18 collectors of 13 different makes have been tested against these standards.
3. In almost all cases, the collectors tested fell short of the proposed standards.
4. The need for adoption of standards for and improvement of design of unit dust collectors is apparent.

APPENDIX

Under controlled conditions, it may be possible to allow recirculation from unit collectors whose effluent does have a higher dust count than the ambient atmosphere without such recirculation by setting up a sliding scale whereby the percentage of recirculation permitted decreases as the effluent dust count increases. A suggested set of standards for this control follows:

1. Maximum permissible recirculation for units having effluent dust count no higher than ambient without recirculation may be 1 cfm for every 10 cu ft of room volume. (This limit is set to provide a factor of safety for improper maintenance, one example of which is the not uncommon practice of removing rather than replacing a dirty filter.)
2. Maximum permissible test dust count above which no recirculation would be permitted should be 3 mppcf in excess of ambient without recirculation when the unit is tested at a dust loading of 1.5 grains per cubic foot by a dust having a mean size of 3μ on a particle count basis.
3. Maximum permissible recirculation for units having effluent dust count of 3 mppcf in excess of ambient air without recirculation under above test conditions should be 1 cfm for every 100 cu ft of room volume.
4. In the intermediate range an additional 10 cu ft of room volume per cfm of air recirculated could be allowed for each step of ⅓ mppcf above ambient without recirculation between 0 and 3 mppcf.

DISCUSSION

J. M. KANE, Louisville, Ky. (WRITTEN): The comprehensive test data presented by Mr. Stern and his associates indicate that unit dust collector designs using an

effective primary dust collector and a high efficiency viscous filter as a final cleaner permit visible and objectionable concentrations of dust to escape to the workroom.

It should be noted that probably 90 per cent of all unit dust collectors are installed to remove dust from metallic grinding, polishing or buffing stands; from tool room machines such as surface, cutter, tool and hob grinders; or from portable or flexible shaft grinders. The dust collected from this group of operations will have distinctly different particle size distribution and higher specific gravity than tumbling mill dust used to secure the test data. Visible escapement of dust is certainly not typical of collector units in this group on such applications and obviously would not be tolerated by users if such visible escapement were existent in actual practice.

While it is agreed that the efficiency of viscous filters on fine particles is lower than on fabric filters of cloth, felt, or similar media, the advantages of high dust holding capacity with slow rise in resistance of the viscous filters should not be overlooked. It is extremely difficult to obtain vibration of fabric filters every 8 hours especially of the small unit collector type. Failure of such maintenance will result in extreme reduction in exhaust volume and will allow dispersion of increasing quantities of dust to the workroom because of insufficient air flow at the exhaust hood.

From the above comments it appears that the rating procedure based on tumbling mill dust or equal is too severe for an estimated 90 per cent of unit dust collector applications and that dust counts on effluent air would be greatly reduced had test conditions reproduced dust loadings typical of that group.

It should also be questioned whether air cleanliness of 3 mppcf (million parts per cubic foot) for effluent air exhausting nontoxic metallic dust should be required, realizing that 5 mppcf is generally accepted as safe limits for 100 per cent free silica, and noting further that unit dust collectors are only applicable to exhaust systems of small air volume. The proposed standards would prevent the installation of a unit collector having a dust count of 5 mppcf (assuming ambient count is 2 mppcf) for a two wheel grinding stand requiring an exhaust volume of 1000 cfm unless the workroom were 250x40x10 ft and further unless this unit dust collector was the only collector in that area.

R. S. FARR, Los Angeles, Calif.: We have experimented in collecting dust by using a precleaner and viscous impingement filter. We found the over-all efficiency of the two units to be less than the efficiency of the filter alone. The precleaner removes the large particles and the viscous filter removes a portion of the small particles that are not collected by the precleaner. The filter itself when collecting large particles as well as small particles together showed higher efficiencies than when the combined unit was tested.

CYRIL TASKER, Cleveland, Ohio: The Society is vitally interested in the subject of air cleaning and as a result of the consideration given this subject at the last meeting of the TAC on Air Cleaning the Research Laboratory will have a fulltime research engineer on air cleaning problems. Arrangements are being made to obtain financial assistance from the air filter industry and it is expected that information obtained will lead to development of a standard code for testing air cleaning devices.

DR. BRANDT: I should like to point out that there are three types of air cleaners or filters. One comprises the large group employed in ventilating systems in office buildings and public buildings, such as stores and theaters, to remove some of the soot and dirt from the air supplied to such occupied spaces. The second group is that which is of particular interest to industrial hygiene engineers; filters or collectors used to remove relatively high concentrations of atmospheric contaminants from the air handled by industrial exhaust systems. The third group consists of those cleaners used on the intakes to compressors and some other types of machines. The cleaners discussed by Mr. Stern fall in the second group.

It is my impression that the Technical Advisory Committee on Air Cleaning is interested only in the first group of cleaners, as K. E. Robinson, Michigan, and I,

were forced to conclude when we attended the meeting of this Committee in January. At least the first two groups of cleaners should be given consideration if we are to make any progress in the industrial hygiene field. The fact that Mr. Stern had to devise his own test procedure is evidence enough that progress in this direction leaves much to be desired.

R. S. DILL, Washington, D. C.: This air filter business has been stalled for 15 years on the matter of standard dust for air cleaner testing purposes. What are you going to use as a test dust and how should you determine what is in the air coming out of the cleaning devices? The main difficulty is that of getting agreement on these and some other points.

I am trying to sell the idea that filters should be tested on a series of dusts of various sizes and characteristics rather than on a single dust for which we talk about the efficiency of various filters. The difference is in the size of the dust. An electrostatic cleaner will capture quite small particles; others, particles not so small. Why not say that one cleaner is good for one size dust and another good for a different size? I hope to see some results from this program which we have tentatively outlined for the Society research. My committee is supposed to work on dust catchers as well as air cleaners.

L. T. AVERY, Cleveland, Ohio: As an air cleaner user I am interested in a rating which the user can depend upon for a continuous performance. Rating in terms of a clean air cleaner seems meaningless.

W. H. CARRIER, Syracuse, N. Y.: It is important to consider the type of dust and the weight per micron as well as the size. For example, there is a great difference in the type of cleaner required for relatively heavy metallic dusts and that required for fine cotton or lint dusts. It is necessary to study the subject broadly in order to decide upon the different types of dust which should be considered as test dusts.

MR. FARR: As a member of this committee, I think we are taking into consideration both of the problems raised by Dr. Carrier and Mr. Avery. One thing that is essential is a standardized test procedure before we can evaluate the effect of different particle sizes; how the change in density of these particles affects the filtering efficiency, and how the concentration of dust also affects the efficiency of the filter. Before we can make any group attempt to evaluate the various variables such as those mentioned we must first have a standard test procedure. When we have a standard test procedure we can evaluate air conditioning filters, electrical filters, and unit dust collectors.

While we all recognize that no one type of filter is a cure-all for all dust problems, we realize that we have to have a standard test procedure that can be used in evaluating different types of units for different types of problems.

S. H. DOWNS, Kalamazoo, Mich.: Discussions of rating usually refer to the rating of complete systems, whereas catalog ratings of air cleaners refer to the performance of an individual part of the system only. The purchaser of a cleaner installs it in a system and then the system design has an effect upon the performance.

AUTHOR'S CLOSURE: The discussers of this paper have, very properly, emphasized the extreme importance of the choice of test dust. Mr. Dill indicates that after more than 15 years of testing there is still disagreement as to the size and other characteristics of a test dust for air filters. It is, therefore, not surprising, that, in this pioneer attempt to rate industrial dust collectors similarly, objection should be raised to our choice of test dust. Mr. Kane's statement that probably 90 per cent of all unit dust collectors are used in metallic grinding and buffing installations is based upon his experience representing a manufacturer who sells directly to industry through its own technically trained sales engineers and, as a result, knows at least the initial application of its portable type equipment. Presumably they would not sell such equipment for an

application where they had doubts of its effective performance. In contrast to this, the great majority of other manufacturers sell unit collectors through jobbers, industrial supply and hardware dealers, and similar channels so that the manufacturers have no knowledge or control of the ultimate sale and application of their units. We have, as a result, seen such equipment in use in a wide variety of applications which no reputable manufacturer would ever recommend for such a product. The same thing has happened when war surplus equipment of all manufacturers has reached plants beyond the manufacturers' control and where equipment originally installed in an application approved by the manufacturer has later been relocated and put to originally unanticipated uses.

It is because of this that we believe that if only one test dust is used it should represent not only the 75 per cent (our figure) to 90 per cent (Mr. Kane's figure) of installations used on metallic grinding or buffing but also the remaining uses which frequently involve dusts much more difficult to collect. A still better procedure would be to rate units against several different test dusts, each representative of certain typical industrial operations such as metallic grinding, buffing, etc. We are currently investigating this approach.

With regard to the suggested standard of air cleanliness, we have had recent additional experience with several cloth bag type unit collectors showing that an effluent dust count of 5 mppcf is obtainable with presently available commercial equipment. It may be, however, that our suggested standard of 3 mppcf above ambient is too low and should be raised upwards.

1297

No. 1297

AIR CONDITIONING A CANADIAN WINDOWLESS TEXTILE MILL

By G. Lorne Wiggs,* Montreal, Que., Canada

THE TREMENDOUS industrial expansion in Canada and the great engineering advances in both the textile machinery and textile mill construction fields in the United States during the war years have resulted in a new trend of textile mill design. Reflecting this recent trend is the new textile mill described in this paper.

The mill is located in the town of Ormstown, Que., situated approximately 40 miles southwest of Montreal. The mill was designed and constructed for producing spun rayon yarns, rayon and wool blends and worsted yarns on the latest type of machinery for handling long staple fiber. It will house a complete unit of 5000 spindles.

A perspective view of the new plant is shown in Fig. 1. It will be seen that the plant consists of a complete spinning mill, approximately 100 ft x 300 ft, which is a single story, completely windowless building, along the front of which are located the general offices, the men's and women's wash rooms, the first aid room, the cafeteria, the boiler room and the machine shop on the ground floor, with the executive offices on the second floor. A penthouse running practically the length of the mill, houses all the air conditioning equipment. A covered loading platform with a railroad siding is also located along a portion of the front of the mill. An electrical substation and an artesian well and pump house are located at the back of the building. Probably the most striking thing about the plant is the small size of the chimney. The design was so made that by constructing additions at each end, the mill could readily be enlarged to increase the unit to one of 10,000 spindles. Further expansion could be made by building a second 10,000 spindle unit along the back of the present unit.

In designing a unit to produce the various yarns mentioned, the prime consideration was to design a building which would not only be an efficient operating unit but one in which the particular air conditions required for spinning each different yarn could be readily obtained at any time. Although the outdoor temperatures in this locality vary from a minimum of about −30 F in winter to a maximum of almost 100 F in summer and there is a very large

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concentrated internal sensible heat gain in the mill, the problem of designing the air conditioning system is not a particularly complex one. The difficulty lies in the determination of the best type of system to be used, in providing the proper air distribution so as to maintain a uniform temperature and humidity throughout the mill without objectionable draughts and in selecting equipment and controls to take care automatically of the widely differing conditions the system must encounter.

There was considerable difficulty in obtaining an authoritative statement of the exact air conditions which should be maintained in the mill for each different type of spinning. It was finally agreed, after consultation with the

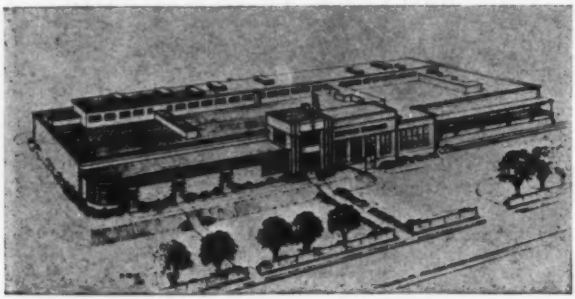

FIG. 1. WINDOWLESS TEXTILE MILL

management and various authorities, that for spinning rayon yarns the air conditions to be maintained should be as follows:

Dry bulb temperature, 75 F to 80 F.
Relative humidity, 60 per cent to 65 per cent.

In spinning rayon and wool blends and spinning worsted yarns on the so-called French spinning system the air conditions required would be as follows:

Dry bulb temperature, 75 F to 80 F.
Relative humidity, 75 per cent to 80 per cent.

Since the mill is initially to spin rayon yarns and since it will probably be some years before rayon and wool blends or worsted yarns will be spun, the management decided to have the plant designed and constructed for the spinning of rayon yarns only, but to design the walls so that at a later date, when other yarns are to be spun, the additional insulation required in the walls to prevent condensation from occurring on these surfaces could be added. These considerations required a thorough study of many different possible types of wall and ceiling construction and of various insulating materials available, so as to produce an economical wall, especially when it is considered that at the time these studies were made many common building materials were extremely scarce in Canada and others were not available at all. The usual type of so called *Mill Construction* could not be adopted because of the shortage of lumber.

Reverting to the air conditions required in the mill when spinning rayon yarns, the following design conditions were adopted for the mill proper:

	SUMMER	WINTER
Outside dry bulb temperature	90 F	−30 F
Outside wet bulb temperature	73 F	
Inside dry bulb temperature	75 F	75 F
Inside relative humidity	65%	65%
Inside wet bulb	66.6 F	66.6 F
Inside dew point	62.5 F	62.5 F

By reference to the graph, Fig. 4, Chapter 6 of THE GUIDE 1946, showing the permissible relative humidities for various transmission coefficients, it will be seen that for an outside temperature of −30 F, an inside temperature of 70 F, and a relative humidity of 65 per cent, the wall, roof or glass heat transmission coefficient *U* must be less than 0.20. The same graph shows that for an outside temperature of −30 F, an inside temperature of 70 F and an inside relative humidity of 80 per cent, the wall, roof or glass coefficient *U* must be less than 0.115. The coefficient *U* of single, double and triple thickness of glass being respectively 1.13, 0.45 and 0.281, it is obvious that for either of the above winter conditions, even if triple thicknesses of glass are used in the windows, condensation will occur on the glass whenever the outside temperature drops below 0 F. This, with the condensation which would occur on the window frames, would present a difficult problem when the relative humidity of the mill was maintained at 65 per cent and would be serious if the relative humidity were raised to 80 per cent. In addition, since the heat transmissions of glass windows or of glass bricks are all so much higher than those of well insulated walls, the omission of all windows materially reduces the cooling load in summer and heating load in winter and at the same time results in more uniform conditions in the mill. Because the omission of windows eliminates a difficult condensation problem and at the same time reduces the size of the air conditioning installation and the heating plant, it was decided to construct the mill proper without windows and with a minimum of exterior doors. However, in order to offset any possible objections that the plant employees might have to work in a completely windowless plant, windows were provided in the offices and service portion of the mill, where condensation is not a problem.

In order to allow for any possible lower temperatures at their inside surfaces, it was decided that the wall coefficient should initially be less than 0.15 and ultimately less than 0.10, while that of the roof should be less than 0.10.

After some study, it was decided to construct the plant with concrete foundations and a single reinforced concrete floor, with a structural steel frame, with exterior walls constructed of white brick bonded with 8 in. terra cotta blocks and with precast light weight aggregate concrete roof slabs, supported on steel bar joists, thus giving a fire resistant construction.

The many different insulating materials commonly used were then investigated. A number of desirable insulating materials were not available at the

TABLE 1—TABULATION OF HEAT GAINS AND HEAT LOSSES

	SUMMER		WINTER			
	Working Day	Week-end	Working Day		Week-end	
	Heat Gain	Heat Gain	Heat Gain	Heat Loss	Heat Gain	Heat Loss
PICKER ROOM:						
Heat Transmission, Solar Heat Gain and Infiltration	9,800	9,800		99,900		99,900
Fan and Duct Heat Gain	8,900	8,900	8,900		8,900	
Internal Heat Gain	29,800		29,800			
			38,700	99,900	8,900	99,900
				38,700		8,900
Net Heat Gain or Loss	48,500	18,700		61,200		91,000
CARDING & DRAWING ROOM:						
Heat Transmission, Solar Heat Gain and Infiltration	78,100	78,100		822,200		822,200
Fan and Duct Heat Gain	25,400	25,400	25,400		25,400	
Internal Heat Gain	188,300		188,300			
			213,700	822,200	25,400	822,200
				213,700		25,400
Net Heat Gain or Loss	291,800	103,500		608,500		796,800
SPINNING ROOM:						
Heat Transmission, Solar Heat Gain and Infiltration	62,600	62,600		656,900		656,900
Fan and Duct Heat Gain	81,100	81,100	81,100		81,100	
Internal Heat Gain	865,500		865,500			
			946,600	656,900	81,100	656,900
			656,900			81,100
Net Heat Gain or Loss	1,009,200	143,700	289,700			575,800

time. Finally it was decided to adopt for the exterior walls a slab insulating material made from shredded wood and cement, which has a conductivity k of 0.46 and which bears the approval of the Underwriters Laboratories. With a 2 in. thickness of this insulating material the value of U for the wall neglecting the interior finish is then approximately 0.14. When, at a later date an additional 2 in. thickness of insulation is added, the heat transmission coefficient U will be less than 0.09.

The insulating board has a rather rough finish, so that in order to obtain a smooth interior finish, it was decided to give the board a coat of neat cement plaster. After this was thoroughly dry, the plaster was given two coats of emulsified asphalt cement, in order to provide a satisfactory vapor barrier. As an additional seal and in order to prevent any possibility of the asphalt

bleeding through the interior paint finish, the emulsified asphalt was given two coats of aluminum paint, which was finally painted with two coats of paint of a light green pastel shade.

The roof was insulated with wood fiber board insulation 2 in. thick which, with the light weight concrete roof slabs, gives U value for the roof of less than 0.10. The inside surface of the roof was sprayed with two coats of emulsified asphalt to provide the necessary vapor seal. Finally, a ceiling of gypsum board was attached to the bottom chords of the roof joists.

Incidentally, the roof of the mill was designed to be dead flat and the rain water leaders have been provided with removable standing waste pipes so that in summer the roof will be flooded with water approximately 2 in. deep.

The mill was designed so that its floor level would be at the level of a standard railway car floor which, due to the fact that the property is practically

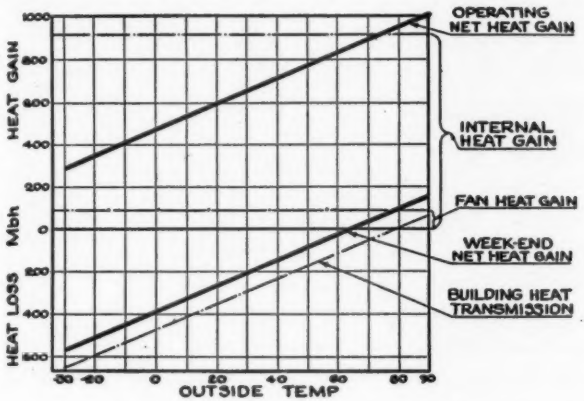

FIG. 2. VARIATION OF HEAT GAINS AND LOSSES OF VARIOUS OUTDOOR TEMPERATURES

flat, meant that the mill floor is roughly 4 ft 5 in. above grade. The plant floor, the walls below floor level and the foundation walls were all constructed of reinforced concrete, the floor being 4 in. thick and the concrete walls 12 in. thick. The amount of exposed concrete wall was reduced by backfilling against the wall on all sides of the plant to a height of 3 ft 10 in. above the grade.

The wall and roof construction of the office and service section of the plant was modified slightly from that used in the mill proper, because the relative humidity to be maintained in this section of the plant is to be that commonly maintained in offices and the like.

The construction of the walls and the roof of the penthouse also are different from that of the plant, but were designed to have the same U value as those of the plant.

The mill proper is without windows of any kind and the only doors in the walls are those between the mill and the offices, one leading to the shipping platform and the regulation number of fire escape or panic doors. The panic

doors were constructed similarly to standard refrigerator doors so as to minimize condensation on them and to prevent the doors from freezing in winter.

Having adopted the foregoing construction for the mill, calculations were then made to determine the various heat gains and heat losses of the mill for both summer and winter conditions.

Table 1 gives a list of the various heat gains and heat losses. It is notable that because of the large internal heat gains in the mill, particularly in the spinning room when it is in operation, even at −30 F, the spinning room requires cooling during the working days. The variation in the heat gains and heat losses in the spinning room at various outdoor temperatures are shown in Fig. 2. This graph is not strictly accurate, but it serves to show the

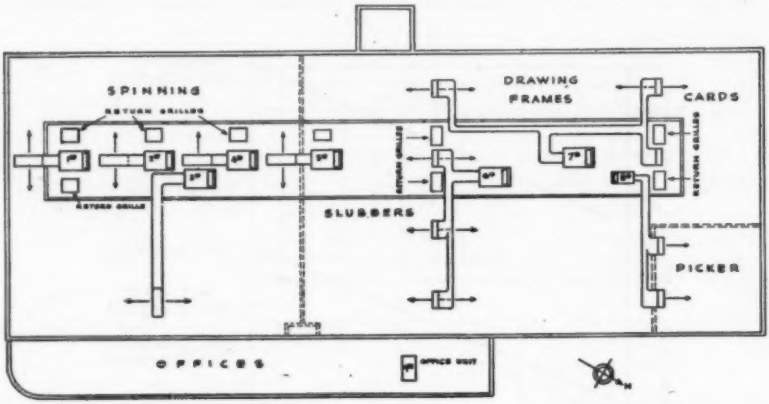

FIG. 3. PLAN OF MILL SHOWING LOCATION OF AIR CONDITIONING UNIT

relation between the various heat gains and heat losses for any given outdoor temperature. It might be mentioned that had the building been constructed with windows, the slope of the inclined lines would have been increased and both the maximum heat gain in summer and the maximum heat loss in winter would have been increased.

With the tabulation of heat gains and heat losses a study was made of the various types of air conditioning systems available, among these being the system having spray atomizers combined with mechanical ventilation, to provide both humidification in winter and evaporative cooling in summer and standard air conditioning systems, both central station type and unit type, to provide year round air conditioning. The unit air conditioning type was finally adopted, because of its flexibility, the ease of its installation and, particularly, because very little of the installation would be exposed in the mill. In order to expedite delivery and to keep costs to a minimum, eight standard unit conditioners were adopted for the mill proper, one being provided for the picker room, two for the carding and drawing room and five for the spinning room. A separate unit was provided for the offices and service areas. All the unit conditioners are factory assembled units, each including standard *capillary* cells, heaters,

fan, pump, motors, etc. The unit conditioners are located as shown in Fig. 3. The outside air inlets to the units are located immediately above the units, while the entire penthouse acts as a return air plenum. The exhaust air is discharged outdoors through the wall of the penthouse. The conditioned air is supplied to the mill through conditioned air inlets of the type shown in Fig. 4, which also shows the continuous lighting troughs and the pendant automatic sprinkler heads. Since the spinning frames will be driven by individual motors, there will be no overhead shafting or belts, with the result that the present clean cut appearance of the mill will be maintained even when the textile machinery is in place.

At the time that the mill was designed all refrigerating equipment was *frozen* in Canada. However, prices were obtained on the required refrigerat-

FIG. 4. SPINNING ROOM BEFORE INSTALLATION OF MACHINERY

ing installation, including evaporative condensers, but at the same time a study was made of the few deep wells in the vicinity and of the geology of the locality. The geological surveys and reports available indicated that the ground in the area, under an over-burden of about 60 ft of clay, had an underlying area of Nepean (Potsdam) sandstone formation of such a character and grain size that it could be expected to yield sufficient water at a reasonably low temperature to cool the plant.

Because of the uncertainty of obtaining refrigeration equipment, and because a source of sufficiently chilled water appeared to be available from a deep well, and because both the operating and maintenance costs of a deep well and of a deep well pump would be materially lower than that of a refrigerating unit, it was decided to sink a deep well 12 in. in diameter. The well was drilled to a depth of about 130 ft, of which about 65 ft is through clay and 65 ft is in bedrock. Later a pumping test of the well was made. After pumping continuously for 24 hours, the official 8 hour test showed a delivery

averaging 827 U. S. gpm with a draw down of less than 20 ft. The temperature of the water was found to be 42½ F and, while its hardness was high, it was otherwise very satisfactory. As a matter of interest, the well drillers reported that they had run the test pump at various capacities to determine the ultimate capacity of the well, but they reached the limit of the capacity of the test pump at about 2500 U. S. gpm without seriously increasing the draw down of the well.

Based on a well water temperature of 45 F the quantity of chilled water required for cooling the plant and offices amounts to a total of 320 U. S. gpm. As a result, the well has not only ample capacity to cool the present mill, but will also be sufficient for cooling the future additions.

The heating of the mill proper will be done by means of steam heating coils in the unit conditioners, while the heating of the offices and service areas

FIG. 5. RADIANT HEATING COILS BEFORE CONCRETE WAS POURED

will be provided by a low temperature radiant panel heating installation in the concrete floors of those areas. The partly fabricated floor coils are shown in Fig. 5. The radiant heating system utilizes forced hot water as a heating medium. In order to prevent the possible occurrence of condensation on the concrete floor of the mill proper, along its junction with the foundation walls, a single pipe coil was built into the junction of the exterior walls and the concrete foundations. The coil is connected to the radiant heating system. A detail showing just how this coil was built into the wall is shown in Fig. 6. The boiler plant consists of two oil fired low pressure steam boilers, which provide steam for the heating of the plant and for heating the domestic hot water. Each boiler has a rated heating capacity of 4500 sq ft EDR when mechanically fired. When the textile machinery is in operation one boiler only will be used, even during the coldest weather, but both boilers will be required during the weekends when subzero temperatures occur. It is due to the fact that the plant is windowless that such a small boiler capacity is required and

that consequently the size of the chimney is so small in comparison to those usually found on mills of similar size. It is planned that at a later date electric heaters will be installed in the conditioned air supply ducts so as to take care of the weekend heating. In this way the steam boilers will be used during the plant operating periods only and the heating during the weekends will be done by electricity purchased as off peak power. Thus the present steam boilers will have sufficient capacity to take care of the heating of the future additions to the mill.

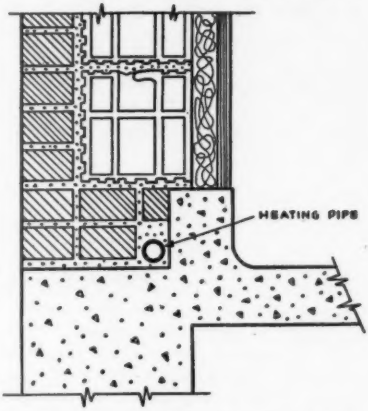

FIG. 6. DETAIL OF SINGLE HEATING COIL IN EXTERIOR WALL

Because so little heating will be required in the mill, the fuel oil burning installation adopted utilizes oil burners approved to burn No. 3 fuel oil.

A schematic diagram of the control system for a typical factory air conditioning unit is shown in Fig. 7. This control system was designed to hold the dry bulb temperature and relative humidity constant the year round, and to control automatically the heating and humidification in winter, the evaporative cooling in spring and fall and the cooling by well water in summer.

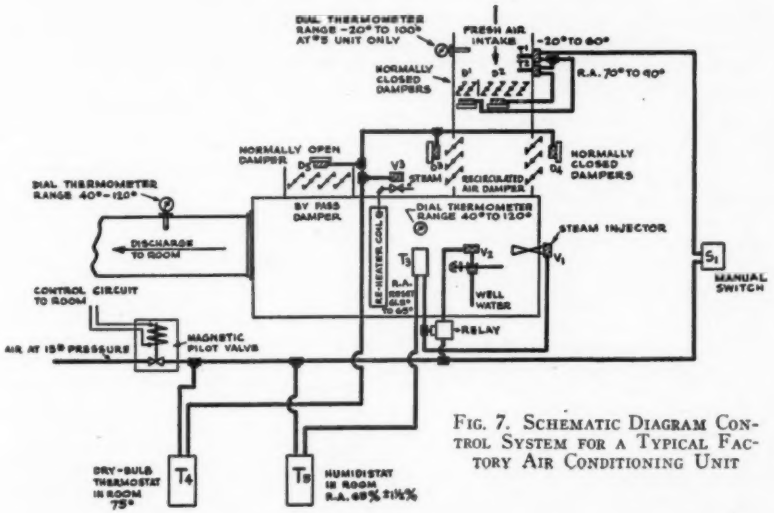

FIG. 7. SCHEMATIC DIAGRAM CONTROL SYSTEM FOR A TYPICAL FACTORY AIR CONDITIONING UNIT

Due to delays caused by difficulty in obtaining materials and by strikes both in the construction industry and, particularly, in the textile machinery manufacturing industry, this mill has not yet been put into production. It is, therefore, impossible at this time to give any data regarding its operation. After the mill has been in operation for some time, it is planned to make a survey of the performance of its air conditioning installation similar to surveys now being made by the writer in a number of other textile mills, and it is expected that the comparative data thus accumulated would form the basis of a subsequent paper.

DISCUSSION

CYRIL TASKER, Cleveland, Ohio: Is the dry bulb temperature of any great significance in the spinning operations outside of its effect on the relative humidity? From the point of operation is not the relative humidity a more important factor? Could you let the dry bulb run up slowly as long as you maintained the right relative humidity? How many operators do you have in a factory of that size? A little while ago I visited a large textile (rayon) plant outside of Cleveland. We walked through a vast factory, and the number of operators present was surprisingly small, so that the human load must be quite a small part of the total load.

L. T. AVERY, Cleveland, Ohio: In order to hold the relative humidity constant, you must hold the temperature constant. What is the construction of the air distribution box with regard to the position of air outlets?

H. E. ZIEL, Detroit, Mich.: Is the fan room construction similar to the first floor side walls and roof, using the same insulating materials? It appears that the same heat transmission factor U would be required for the walls and roof of the fan room as the first floor air is drawn into the fan room.

Was a closed type exhaust and recirculating system considered in place of using the fan room for an exhaust and recirculating chamber? A closed type system would reduce the need for insulating the fan room walls and roof.

R. S. DILL, Washington, D. C.: If cleaners were used in the re-circulating air stream, I would like to know what kind were used.

MR. WIGGS: In answer to Mr. Tasker's question, I believe that most mill operators desire constant relative humidity rather than constant dry bulb temperature, but for the production of cotton yarns and fabrics, I am not certain whether the relative humidity or the absolute humidity is the more important.

With regard to the number of operators I would say that this mill has a total of about 100 operators and maintenance men, divided into three shifts.

The air distribution boxes, Mr. Avery mentioned, were designed for streamlined flow from the units into the box and have grille outlets on both sides of the box, with each outlet grille equipped with both vertical and horizontal adjustable vanes.

Answering Mr. Ziel's question, the coefficient of heat transmission U of the walls of the fan room is the same as that of the walls of the mill proper, but the actual construction is totally different.

The air cleaners in the plant, to which Mr. Dill refers, consist of wire mesh lint arresters in the return air outlets, the air washers and filters in the by-passes around the air washers. The spinning frames and the rewinders are equipped with travelling air cleaners which blow the lint off the machines to the floor where a central vacuum cleaning system is used to pick up the lint.

1298

No. 1298

THE SOL-AIR THERMOMETER—A NEW INSTRUMENT

By C. O. Mackey* and L. T. Wright, Jr.,** Ithaca, N. Y.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with Cornell University.

Sol-Air Temperature

THE CONCEPT and the significance of sol-air temperature in the application of weather data to the estimate of summer cooling loads have been presented in three earlier papers.[1] Briefly, the sol-air temperature combines the effects of air temperature in the shade, solar radiation incident upon a building surface or net radiation emitted by that surface to a cold sky, absorptivity of the surface and wind movement. Between sunrise and sunset, when solar radiation is incident upon the surface of a building material which does directly transmit solar radiation, the sol-air temperature (t_e) for that surface is

$$t_e = t_a + \frac{bI}{h},$$

where

t_a = temperature of the air in the shade, Fahrenheit;
I = intensity of total solar radiation (direct and diffuse) incident upon the surface, Btu per (hr) (sq ft);

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[1] A.S.H.V.E. Research Report No. 1255—Periodic Heat Flow—Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. Transactions, Vol. 50, 1944, p. 293.) Summer Weather Data and Sol-Air Temperature—Study of Data for New York City, by C. O. Mackey and E. B. Watson. (A.S.H.V.E. Transactions, Vol. 51, 1945, p. 75.) Research Report No. 1268—Summer Weather Data and Sol-Air Temperature—Study of Data for Lincoln, Nebraska, by C. O. Mackey. (A.S.H.V.E. Transactions, Vol. 51, 1945, p. 93.)

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Montreal, Que., Canada, June, 1946.

b = absorptivity of the surface for such solar radiation (with values from 0 to 1);
h = outdoor air film coefficient of heat transfer by convection, Btu per (hr) (sq ft) (Fahrenheit degree)

The instantaneous rate of entry of heat into the outer surface of the material with either steady or unsteady heat flow is

$$\left(\frac{q}{A}\right)_L = h(t_e - t_L),$$

where

t_L = temperature of the outer surface of the material, Fahrenheit.

For example, if $t_a = 90$ F, $b = 0.7$, $I = 200$ and $h = 4$, $t_e =$ 125 F and the rate of entry of heat into the surface of the material is precisely as if the air temperature were 125 F with no radiation incident upon the surface.

Between sunset and sunrise when there may be a net transfer of radiant energy *from* the surface *to* a cold sky, the sol-air temperature is

$$t_e = t_a - \frac{I'}{h},$$

where

I' = the net transfer of radiant energy *from* the surface to surroundings, Btu per (hr) (sq ft)

For example, if $t_a = 70$ F, $I' = 30$ and $h = 3$, $t_e = 60$ F and the rate of heat transfer into the surface (or away from the surface, if t_L is greater than 60 F) is precisely as if the air temperature were 60 F with no radiant heat exchange.

THE SOL-AIR THERMOMETER

When Weather Bureau data are used to evaluate sol-air temperatures, there are many gaps in the available information. Temperatures of the air in the shade are measured hourly; at some stations, the intensity of total solar and sky radiation incident upon the horizontal is measured; wind movement is measured. Intensities of total solar and sky radiation incident upon vertical surfaces of various orientations are *not* measured; radiant heat exchange with the cold sky at night is *not* measured; there is no attempt, of course, to measure solar absorptivities or film coefficients of convective heat transfer of various surfaces.

Lacking such data, there is an obvious need for a meter which will measure, directly, the sol-air temperature day or night. The instrument shown in Fig. 1 was devised for this purpose. The meter may have many different forms, but there are certain principles to be followed in its construction. For any exterior surface which does not directly transmit solar radiation, the sol-air temperature is the actual temperature of that surface when it is receiving solar radiation or emitting radiation to a cold sky with convective transfer of heat to or from the surrounding air, only when there is no transfer of heat *from* that surface to the interior or no transfer of heat *to* that surface from the interior.

The instrument designed is in the form of a cube; resting upon the base at the center of the cube is an 8 in. cube of cork. The horizontal and four vertical faces of this cube are faced with aluminum foil; four curtains of aluminum foil are then hung from frames as shown to create four air spaces of ¼ in. width on each face. In this way, considerable thermal resistance is placed in the path of heat flow between exterior faces of the instrument. This puts each exterior face of the meter in heat exchange relationship with its exterior surroundings, only. With this arrangement, the exterior faces of the

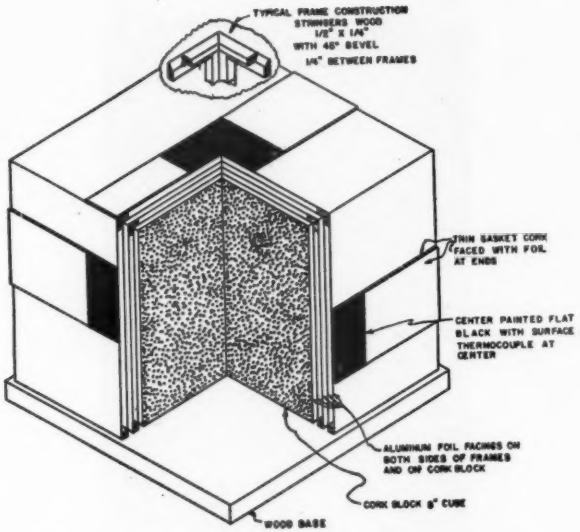

FIG. 1. SOL-AIR THERMOMETER CONSTRUCTION DETAILS

meter will assume and maintain their respective sol-air temperatures. It should be emphasized that the surfaces of a building do not assume the sol-air temperature, commonly, because of conduction of heat away from these surfaces toward the interior or from the interior toward the surfaces. The temperature indicated by the meter is not the temperature that would be assumed by a building surface, but is very close to the true sol-air temperature.

Another principle of design is that the surface of the meter should have low heat capacity in order that the temperature of the surface may respond quickly to changes in the external conditions of solar radiation, air movement, or air temperature. In the meter shown, the strips of thin gasket cork were fastened to the aluminum foil on the four vertical faces and on the top horizontal face; cork has extremely low volumetric heat capacity. A thermocouple of No. 36 copper and constantan wires was imbedded in each cork strip and the leads

were carried in this cork for about 2 in. Five leads were then carried to a selector switch from the cork. A copper constantan couple to measure air temperature was hung below the instrument in the shade of the base and leads from this couple were also carried to the selector switch. In this way six temperatures may be measured with a potentiometer.

In this first instrument, a small square of the cork containing the imbedded thermocouple was painted with two coats of No. 42 Sherwin and Williams black paint which has an absorptivity approaching unity. Sol-air temperatures

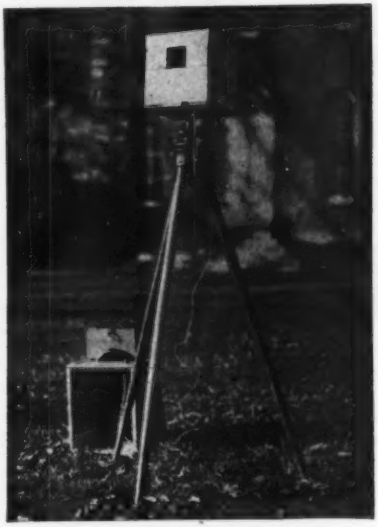

FIG. 2. SOL-AIR THERMOMETER MOUNTED ON TRIPOD

measured by this instrument are, therefore, the sol-air temperatures of surfaces with an absorptivity of substantially 1.0.

One of the features of this instrument, however, is that the sol-air temperatures for surfaces of different solar absorptivities may be found without actually determining these absorptivities. Thin slices of the materials in question may be affixed to the external foil and the thermocouples imbedded in these surfaces; the temperature assumed is then the sol-air temperature of that particular surface under the existing external conditions. Thus, thin slices of wood may be painted different colors and a thermocouple imbedded in each different patch will record a sol-air temperature for a wood surface painted with that color; thin slices of brick or mortar may be similarly attached; thin pieces of roofing materials may also be investigated in the same manner.

This procedure also gives a method of determining solar absorptivities by proper comparison of the sol-air temperature of any surface with the sol-air temperature of a *black* surface of the same orientation; the air temperature is the only other variable that must be measured for this purpose.

The cube shown was mounted upon the tripod of a surveyor's level and two levels were attached at right angles so that the top surface could be placed in a true horizontal position and the four vertical faces in true vertical positions. A compass was included in the mounting so that the orientation of

FIG. 3. SOL-AIR THERMOMETER MOUNTED ON STAND

the vertical surfaces might be known. Photographs of the meter are shown in Figs. 2, 3 and 4.

DAYTIME DATA

The meter was not completed until the fall of 1945. On a clear day, October 11, 1945, the meter was set up on the Cornell campus at Ithaca, N. Y. The vertical surfaces of the cube were oriented to face the true north, east, south and west. Temperatures of the five faces of the cube and of the air in the shade were recorded with the results as shown in Table 1.

The scale of millivolts on the potentiometer used was such that the observed temperatures are probably accurate to ±2 F.

These readings show that the meter is successful in maintaining a large temperature difference between the several faces and they also indicate that

the response of the meter to changing external conditions is rapid. Observers noted a large and rapid change of sol-air temperature with changes in wind direction and speed. The readings indicate the complex nature of heat transfer in buildings caused by rapidly changing ambient conditions. For example, at noon on the day of the test, the external surface of a building with a solar absorptivity of nearly one would be *gaining* heat as if exposed to an air temperature of 92 F if horizontal and 112 F if facing south, while it might be *losing* heat as if exposed to an air temperature of 63 F if facing east, of 61 F if facing north and of 57 F if facing west; all this occurs with an actual temperature of the air in the shade of 50 F. There is a sol-air temperature

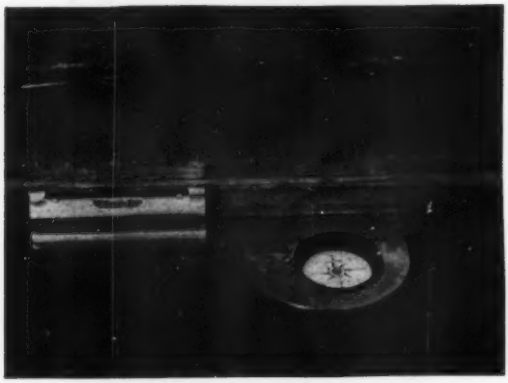

FIG. 4. SOL-AIR THERMOMETER—CLOSE-UP VIEW OF LEVEL AND COMPASS

for the human body, of course; on the day of the test, students without topcoats were comfortable with an air temperature of only 50 F.

NIGHT SOL-AIR TEMPERATURES

The question of the effect of night radiation from building surfaces to a cold sky upon rate of heat transfer during the following day has been raised in several discussions of the three papers previously mentioned. Since the Weather Bureau data contained no information on this question, night sol-air temperatures for New York, N. Y., and for Lincoln, Neb., were taken equal to the observed air temperatures.

Actually, during the night hours, if the angle factor from surface to sky is unity (if the building surface *sees* only sky), the sol-air temperature of the surface is

$$t_o = t_a - \frac{\epsilon I_a}{h},$$

TABLE 1—TOP AND SIDE TEMPERATURES OF SOL-AIR THERMOMETER ON OCTOBER 11, 1945, ITHACA, N. Y.

EASTERN STANDARD TIME	AIR TEMPERATURE IN SHADE, F	OBSERVED SOL-AIR TEMPERATURES FOR SOLAR ABSORPTIVITY OF 1.0, F				
		Horizontal Surface	Vertical Surface Facing			
			North	East	South	West
11:30 A.M.	50	95	61	66	108	59
11:45	49	89	58	65	109	55
12:00 NOON	50	92	61	63	112	57
12:15 P.M.	52	97	64	63	121	66
12:30	50	88	61	59	115	66
12:45	52	92	62	59	112	73
1:00	51	87	61	61	110	78
1:15	52	90	61	56	109	82
1:30	51	88	60	57	105	84
1:45	53	94	66	65	111	94
2:00	54	88	66	61	102	92
2:15	53	86	63	61	106	96
2:30	53	82	62	61	102	97
2:45	54	79	63	61	98	97
3:00	53	76	62	59	94	101
3:15	55	76	62	61	93	107
3:30	57	74	62	61	91	101
3:45*	56	56	56	56	57	62

* Sun went under a cloud at this time.

where

e = emissivity of the surface;
I_n = net rate of heat transfer by radiation from a *black* surface to the cold sky, Btu per (hr) (sq ft);

$$= 0.173\left[\left(\frac{T_L}{100}\right)^4 - \left(\frac{T_s}{100}\right)^4\right]$$

where

T_L = the absolute temperature of the surface;
T_s = the effective radiating absolute temperature of the sky.

For the expected temperature of surface and sky on *clear* nights,

$$I_n = 0.68(t_L - t_s) \text{ (approximately)}$$

In order to estimate the probable effect of radiation to a *clear* sky during the night, it may be assumed that the effective radiating temperature of a clear sky is about −60 F. For a horizontal roof located in New York, N. Y., design sol-air temperatures shown in Table 2 have been calculated. The surface has been assumed to have a solar absorptivity of 0.7 and an emissivity for radiation to a cold sky of 1.0; it is assumed that the horizontal building surface *sees* only a *clear* sky at an effective radiating temperature of −60 F between sunset and sunrise. Design sol-air temperatures are given in Table 2 for each hour of the day, with and without radiation to the sky at night.

As an example of the effect upon cooling load of this radiation to a *clear* sky at night, precise calculations have been made for one roof slab. The roof is assumed to be homogeneous with a thickness of 6 in.; the material chosen is concrete with $k = 1$ Btu per (hr) (sq ft) (Fahrenheit degree per ft) $\rho = 143$ lb per cu ft, and $c = 0.21$ Btu per (lb) (Fahrenheit degree). The maximum contribution to the cooling load due to heat transfer through this roof would

TABLE 2—CALCULATED DESIGN SOL-AIR TEMPERATURE FOR A HORIZONTAL ROOF IN NEW YORK, N. Y.

HOURS AFTER NOON	SOL-AIR TEMPERATURE, F		HOURS AFTER NOON	SOL-AIR TEMPERATURE, F	
	No Night Radiation	Night Radiation to Sky at − 60 F		No Night Radiation	Night Radiation to Sky at −60 F
0 (NOON)	135	135	13	78	58
1	136	136	14	77	57
2	134	134	15	77	57
3	129	129	16	76	56
4	123	123	17	76	56
5	112	112	18	79	79
6	101	101	19	92	92
7	91	91	20	102	102
8	85	64	21	114	114
9	83	62	22	122	122
10	82	61	23	130	130
11	81	60			
12 (MID.)	79	59	Avg.	99.8	91.2

TABLE 3—EFFECT OF RADIATION TO A CLEAR SKY UPON COOLING LOAD

	NO RADIATION TO NIGHT SKY	NIGHT RADIATION TO SKY AT −60 F
Calculated temperature of inside surface of slab, F	99.3	97.4
Calculated rate of heat transfer from slab to indoor air, Btu per (hr) (sq ft)	31.8	28.7

occur at about 4 p.m. At this hour and for the design sol-air temperatures previously reported, the temperatures of the inside surface of the roof and the corresponding rates of heat transfer to indoor air at 80 F with and without an allowance for radiation to a clear night sky are shown in Table 3.

The conclusion is that radiation to a sky at −60 F between sunset and sunrise causes a 10 per cent lower rate in the *maximum* contribution to the cooling load from a horizontal 6 in. concrete roof slab in New York, N. Y., as compared with the maximum load when no night radiant exchange is assumed to occur. In the example, a clear sky at −60 F was assumed to exist at all times between sunset and sunrise; this gives rise to a probable *maximum* reduction in cooling load, for there is no assurance that such a situation would

exist in combination with the high daytime sol-air temperatures. The effect of night radiation upon cooling load would be still less for vertical walls, because they exchange heat by radiation at night not only with the sky but with surfaces at much higher temperatures such as pavement, ground, trees and the surfaces of other buildings. In other words, the night sol-air temperatures for vertical surfaces would be expected to be higher than for horizontal surfaces.

NIGHT DATA

The sol-air thermometer was set up on only one night in Ithaca, N. Y., October 22, 1945, with the readings shown in Table 4.

TABLE 4—TOP AND SIDE TEMPERATURES OF SOL-AIR THERMOMETER ON NIGHT OF OCTOBER 22, 1945, ITHACA, N. Y.

EASTERN STANDARD TIME	AIR TEMPERATURE IN SHADE, F	OBSERVED SOL-AIR TEMPERATURES FOR EMISSIVITY OF 1.0, F				
		Horizontal Surface	Vertical Surface Facing			
			North	East	South	West
12:30 A.M.	42	33	38	39	39	39
12:45 A.M.	42	32	37	38	38	38
1:00 A.M.	41	31	36	37	37	37

The night in question was clear with a full moon. As explained in connection with the daytime readings, these temperatures are accurate only to ±2 F. As expected, the night sol-air temperature for the horizontal surface was observed to be lower than for vertical surfaces. The maximum difference between the air temperature and the sol-air temperature for the horizontal surface in these few observations was 10 F.

SUMMARY

The authors believe that further progress in the analysis of summer weather data, insofar as they may be applied to the estimate of cooling loads, is handicapped by an incomplete knowledge of: (1) solar absorptivities of surfaces of various building materials; (2) effect of wind velocity and surface condition upon exterior convective film coefficient of heat transfer; (3) intensity of solar radiation incident upon *vertical* surfaces; (4) radiant heat exchange between building surfaces and the cold night sky.

The authors believe that the sol-air temperature concept is the best method of combining the several variables in their effect upon cooling load. They believe that a meter which measures sol-air temperature, directly, is superior to any attempt to determine, separately, the several variables and combine them by calculation. Accordingly, they have submitted a design for such a sol-air thermometer. One such meter has been built and a few observations from its use are included.

The authors recommend that meters of this or improved design be used in several key cities in order to obtain more accurate information concerning summer sol-air temperatures.

The authors also recommend a program directed toward the improvement and calibration of the sol-air thermometer as a field instrument. The principal uses of the final meter should be: (1) to determine design sol-air temperature *vs.* time cycles for several different localities; (2) to determine the solar absorptivities of the surfaces of various important building materials.

DISCUSSION

IRVING F. HAND, Milton, Mass.[2] (WRITTEN): This paper describing the sol-air thermometer is most interesting. We note, however, that the authors when referring to Weather Bureau data, state, "Intensities of total solar and sky radiation incident upon vertical surfaces are not measured." We have been obtaining measurements of solar, sky and ground radiation impinging upon vertical surfaces facing south and east for over a year, but we are not surprised that the authors were unaware of this work. Our first public mention of this fact was made at the meetings of the *American Meteorological Society* in Washington, May, 1946.

Among the various types of solar measurements now made by this bureau and cooperating institutions are:

1. Total solar and sky radiation received on a horizontal surface at over 20 stations in continental United States and Alaska.

2. Normal incidence, or direct solar radiation, at Blue Hill Observatory, Milton, Mass., Boston, Mass., Madison, Wis., Lincoln, Neb., and Albuquerque, New Mex.

3. Diffuse radiation received on a horizontal surface at Blue Hill Observatory. For this type of measurement we use an occulting ring, which through adjustment every few days, continuously shades the receiving surface of a 50-junction Eppley 180 deg pyrheliometer from the direct rays of the sun. Following tests, we adjusted the reduction factor to allow for the small portion of sky-radiation masked by the remaining portion of the arc of the occulting ring.

4. Measurements of the total solar, sky and ground radiation received on vertical surfaces facing south and east.

The Weather Bureau also has made several series of nocturnal radiation measurements.[3] Difficulties of retaining an efficient surface have so far prevented continuous measurements of outgoing radiation. We cannot use rock-salt windows for obvious reasons, while the use of sylvite and other infra-red transmitting materials is impracticable at this time. We have found that bare receiving surfaces give inconsistent results; possibly the authors have overcome that difficulty. In fact, the whole problem of obtaining a high degree of accuracy has been studied rather thoroughly by this bureau. For example our Instrument Division has two Eppley pyrheliometers constructed with additional thermocouples within so that the reaction could be watched from many angles. A report recently submitted by the U.S. National Bureau of Standards is most encouraging for it shows that the errors arising from temperature fluctuations and other causes are far less than we had supposed. Our instruments

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[2] Official in Charge, Solar Radiation Section, U. S. Weather Bureau, Milton, Mass.

[3] Monthly Weather Review, 46:57-70, 1918. Transactions, American Geophysical Union, 1937, Part 1, pp. 127-130. Monthly Weather Review, 48:461-462, 1920.

for measuring solar radiation at normal incidence are even more precise than those for measuring total solar and sky radiation. The Smithsonian Silver-Disk pyrheliometer, for example, may be read time after time by two observers simultaneously with differences amounting to less than one-half per cent, while really expert observers can read them within one-quarter of one per cent. Tests with a new Eppley thermoelectric pyrheliometer, specially made and protected by considerable insulation and made waterproof, show a variation not exceeding one per cent throughout the year.

Some work studying albedos has been done by our bureau. Most of this work, however, was for the U.S. Quartermaster Corps and is not available for public distribution. Considerable information on this subject will be found in Smithsonian Tables and Bureau of Standards publications. We thoroughly realize the need for data of this type and should sufficient funds be appropriated for the purpose, we hope to extend the program of measurements of radiation received on vertical surfaces and also albedo measurements of various building materials.

The matter of *film coefficients of convective heat-transfer,* is a subject entirely foreign to the Weather Bureau and we doubt if we could justify studying this problem.

We hope that the authors again will give some of the results obtained with the sol-air thermometer, for this instrument shows considerable promise as a meter which will measure directly, the sol-air temperature day or night. We also hope that some of our measurements of radiation received on vertical surfaces might be utilized in further tests. A paper entitled *Preliminary Measurements of Solar Energy Received on Vertical Surfaces* appeared in the October, 1947, issue of the *Transactions* of the *American Geophysical Union.* A paper entitled *Solar Energy for House Heating* appeared in the December, 1947, issue of *Heating & Ventilating.*

C. F. KAYAN, New York: I should like to congratulate the authors on developing an instrument which gives promise of much future use. Were any temperature measurements made at the core of the instrument and how were the outside surface coefficients determined for different air velocities?

C. M. ASHLEY, Syracuse, N. Y.: Professors Mackey and Wright are to be congratulated on an excellent piece of work, and I can foresee many interesting and valuable uses for this new instrument. Among the practical problems which must be worked out in connection with the determination of the accuracy and scope of the instrument are not only the determination of the effect of size of the instrument on the readings, but also the effect of various wind directions. It should also be pointed out that large size of the instrument itself may not be sufficient, and it may also be necessary to consider a large size panel as well. Another means of check of the instrument would be to compare readings of the instrument with readings of insulated panels placed on the sides of a building of considerable size.

It is to be hoped that through the accumulation of data using this instrument, it will be possible to combine the effects of intensity of sunlight radiation, outside temperature, and wind velocity, and correlate these with outside wet and dry bulb temperature.

G. L. TUVE, Cleveland, Ohio: Has any application of the instrument been made to determine emissivity?

L. T. AVERY, Cleveland, Ohio: Could development or modification of this instrument be used for inside work? I have felt for some time that we needed a combination instrument to measure temperature, radiant heat, air motion and humidity, in order to determine the combined condition around an industrial worker. Could this instrument be used for such a purpose?

J. N. LIVERMORE, Detroit, Mich.: On two phases of the practical application of this instrument I am not entirely clear. Since the author states that the success of

this instrument is dependent upon having surface materials of low specific heat and small thermal lag, can reliable results be expected when thin samples of materials of high specific heat are cemented to its exterior? Is the author satisfied that it is practicable to obtain a surface layer of such materials as brick and mortar thin enough for test purposes?

R. S. DILL, Washington, D. C.: Is it not a fact that the problem is to relate mathematically the indications of this instrument with the heat gain of buildings so that the ultimate form of the instrument is not very important so long as it is standard?

AUTHORS' CLOSURE: No measurements of temperature were made at the core of the instruments. Film coefficients of convective heat transfer were not measured, nor were wind velocities. The readings of the instrument are responsive to air temperature, incident radiation and air movement.

Since the initial use of the instrument, thin samples of painted wood have been attached to the surfaces. The readings obtained indicate that the instrument may be used for the measurement of absorptivities of surfaces for solar radiation in the following manner: (1) attach reference sample and surface under test to the same face of the instrument with each surface then exposed to the same radiation and the same air movement; (2) read sol-air temperature for reference surface (t_o) and sol-air temperature for test surface (t_s); (3) read air temperature in the shade (t_a). Then, the solar absorptivity of the test surface is

$$b = b_o \left(\frac{t_s - t_a}{t_o - t_a} \right)$$

where b_o is the solar absorptivity of the reference sample.

The authors hope that others may be sufficiently interested in the potentialities of this instrument to build more sol-air thermometers of improved design. In its present form, the instrument has no great application to problems of indoor comfort.

1299

No. 1299

PERIODIC HEAT FLOW—COMPOSITE WALLS OR ROOFS

By C. O. MACKEY* AND L. T. WRIGHT, JR.,** ITHACA, NEW YORK

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Cornell University.

THIS paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Cornell University. The study is an extension of the work previously done on heat flow through homogeneous materials and reported in another paper.[1] The contents of this paper in slightly different form have been presented as a report to the A.S.H.V.E. Research Technical Advisory Committee on Cooling Load in Summer Air Conditioning.

COMPOSITE CONSTRUCTIONS

The type of composite construction with which the present discussion is primarily concerned might be described as a *sandwich* type. The wall or roof is composed of two or more layers arranged so that heat flows through the layers in series. Each layer itself is assumed to be a homogeneous material making perfect thermal contact with each adjacent layer.

The basic problem is to determine the temperature of the inside surface of the wall or roof at any time for the case where (1) the incident solar radiation and temperature of the outdoor air are periodic, (2) the temperature of the indoor air is constant, (3) the outdoor air film coefficient of heat transfer is 4.0 and (4) the indoor air film coefficient of heat transfer is 1.65 Btu/hr ft² F.

EQUIVALENT HOMOGENEOUS CONSTRUCTION

The basic problem has been solved mathematically for two and three layer constructions and the important equations are given in a more complete paper

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[1] A.S.H.V.E. RESEARCH REPORT No. 1255—Periodic Heat Flow—Homogeneous Walls or Roofs by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 293.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Montreal, Que., Canada, June, 1946.

of which this is an abbreviated form. In order to make the results of this mathematical solution more practical, the solution has been used to define an *equivalent homogeneous construction. The equivalent homogeneous construction is here defined as a simple homogeneous wall or roof which will have the same variation in indoor surface temperature with time as does the actual composite wall or roof under identical ambient conditions.* The rate of heat transfer from the inside surface to the room air is dependent on the temperature of the inside surface and the inside film coefficient of heat transfer. Thus, the composite construction and its equivalent homogeneous construction will give the same contribution to the cooling load at the same time. It is true that the temperature gradients will be different in the composite and homogeneous constructions, but this is unimportant in finding the cooling load. Also the equivalent construction defined in this paper is the equivalent *only in the particular case of periodic heat flow treated in this study and is not the equivalent construction for all other possible cases of unsteady or transient heat flow.*

Two thermal properties are necessary to define the equivalent homogeneous construction, the thermal resistance $\left(\frac{L}{k}\right)_s$ and the quantity $(k\rho c)_s$,

where

L = thickness of the material, feet;
k = thermal conductivity, Btu/hr ft F;
ρc = volumetric specific heat, Btu/ft³ F.
Subscript s denotes equivalent homogeneous construction.

The thermal resistance of the homogeneous material is determined by the fact that the steady flow characteristics of both the composite and homogeneous constructions must be the same. Let the material of the exterior surface of the wall or roof be designated by the subscript L, and the material of the inside (room side) surface of the wall or roof be designated by o. The materials of interior, central layers of the composite wall or roof will be designated by m_1, m_2, etc.

For a multilayer, composite construction, the thermal resistance of the equivalent homogeneous construction is

$$\left(\frac{L}{k}\right)_s = \left(\frac{L}{k}\right)_o + \left(\frac{L}{k}\right)_L + \left(\frac{L}{k}\right)_{m1} + \left(\frac{L}{k}\right)_{m2} + \cdots \quad (1)$$

If the construction contains an air space, it is recommended that the thermal resistance of the air space be used for $\left(\frac{L}{k}\right)_m$ in Equation 1. Since the mean temperature of this air space will commonly be above 80 F for summer conditions, a reasonable value of thermal resistance for air spaces of 1 in. or more is 0.75 hr ft²F/Btu.

The property $(k\rho c)_s$ of the equivalent homogeneous wall or roof is determined from an empirical equation derived to fit the results obtained from a

number of cases solved by the rational, complete equations which have been derived for this case of unsteady heat flow.

For a multilayer composite construction, the property $k\rho c$ of the equivalent, homogeneous construction is

$$(k\rho c)_e = \frac{1.1\left(\frac{L}{k}\right)_o (k\rho c)_o + 1.1\left(\frac{L}{k}\right)_{m1} k\rho c_{m1} + 1.1\left(\frac{L}{k}\right)_{m2} (k\rho c)_{m2} + \cdots}{\left(\frac{L}{k}\right)_e}$$

$$+ \frac{(k\rho c)_L\left[\left(\frac{L}{k}\right)_L - 0.1\left(\frac{L}{k}\right)_o - 0.1\left(\frac{L}{k}\right)_{m1} - 0.1\left(\frac{L}{k}\right)_{m2} - \cdots\right]}{\left(\frac{L}{k}\right)_e} \quad . \quad . \quad (2)$$

In certain cases, the multiplier (in brackets) of the term $(k\rho c)_L$ in Equation 2 becomes negative; when this occurs, the entire product should be called zero. This negative value is encountered when the thermal resistance of the outside layer is small in comparison with the thermal resistances of the other layers.

Where one of the layers is air (air space), the $(k\rho c)_m$ of the air space may be treated as zero.

TABLE 1—DETERMINATION OF PROPERTIES OF A SIMPLE HOMOGENEOUS WALL OR ROOF, EQUIVALENT TO THE ACTUAL COMPOSITE WALL OR ROOF

CONSTRUCTION	LAYER	SYMBOL	MATERIAL	THERMAL CONDUCTIVITY k BTU/HR FT F	THICKNESS		THERMAL RESISTANCE, L/k HR FT² F/BTU	$k\rho c$ BTU²/HR FT⁴ F²
					In.	L FT		
Wall A	Outside	L	Concrete	1.0	8	0.667	0.667	35
	Inside	o	Corkboard	0.025	3	0.25	10.0	0.10
Wall B	Outside	L	Face Brick	0.767	4	0.333	0.44	20
	Central	m_1	Common Brick	0.417	4	0.333	0.80	8.3
	Central	m_2	Air Space				0.75	0
	Inside	o	Plaster on Metal Lath				0.23	5.4
Roof C	Outside	L	Built up Roofing	0.11	0.375	0.0312	0.28	1.2
	Central	m_1	Insulating Board	0.028	1	0.0833	3.0	0.15
	Inside	o	Concrete	1.0	4	0.333	0.333	35
Roof D	Outside	L	Built up Roofing	0.11	0.375	0.0312	0.28	1.2
	Central	m_1	Insulating Board	0.028	1	0.0833	3.0	0.15
	Central	m_2	Concrete	1.0	4	0.333	0.333	35
	Central	m_3	Air Space				0.75	0
	Inside	o	Plaster on Metal Lath				0.23	5.4

EXAMPLES

Some examples will be given to illustrate the determination of the properties of a simple, homogeneous wall or roof which is the equivalent of the actual, composite wall or roof (see Table 1).

The simple, homogeneous equivalent of Wall A has a thermal resistance (from Equation 1) of

$$\left(\frac{L}{k}\right)_e = 10 + 0.667 = 10.667 \text{ hr ft}^2 \text{ F/Btu},$$

and an equivalent $k\rho c$ property (from Equation 2) of

$$(k\rho c)_e = \frac{1.1(10)(0.1)}{10.667} = 0.103 \text{ Btu}^2/\text{hr ft}^4 \text{ F}^2$$

Note that the last term in the numerator of Equation 2 has been ignored because it is negative.

For Wall B, the simple, homogeneous equivalent wall has a thermal resistance (from Equation 1) of

$$\left(\frac{L}{k}\right)_e = 0.23 + 0.44 + 0.80 + 0.75 = 2.22,$$

and an equivalent $k\rho c$ property (from Equation 2) of

$$(k\rho c)_e = \frac{1.1(0.23)(5.4)+1.1(0.8)(8.3)+20(0.44-0.023-0.08-0.075)}{2.22} = 6.27$$

For Roof C, the simple, homogeneous equivalent roof has a thermal resistance (from Equation 1) of

$$\left(\frac{L}{k}\right)_e = 0.333 + 0.28 + 3.0 = 3.613,$$

and an equivalent $k\rho c$ property (from Equation 2) of

$$(k\rho c)_e = \frac{1.1(0.333)(35) + 1.1(3.0)(0.15) + 1.2(0.28 - 0.0333 - 0.30)}{3.613} = 3.69$$

The simple, homogeneous equivalent of Roof D has a thermal resistance (from Equation 1) of

$$\left(\frac{L}{k}\right)_e = 0.23 + 0.28 + 3.0 + 0.333 + 0.75 = 4.593,$$

and an equivalent $k\rho c$ property (from Equation 2) of

$$(k\rho c)_e = \frac{1.1(0.23)(5.4) + 1.1(3.0)(0.15) + 1.1(0.333)(35)}{4.593} + \frac{1.2(0.28 - 0.023 - 0.30 - 0.0333 - 0.075)}{4.593} = 3.21$$

THERMAL PROPERTIES AND APPARENT DENSITIES OF BUILDING MATERIALS

The apparent density of a building material is extremely important in determining the properties that affect its thermal behavior. For densities above about 5 lb per cubic foot, the thermal conductivity of a material generally increases as the density increases, regardless of the origin of the material. The volumetric specific heat is the product of the gravimetric specific heat and the apparent density. There is not a wide range in gravimetric specific heats of non-metallic solids; solids of mineral origin have gravimetric specific heats between about 0.15 and 0.25 Btu/lb F, while the specific heats of *dry* solids of vegetable origin are commonly between 0.3 and 0.4 Btu/lb F at temperatures important in this study. Consequently, the volumetric specific heat of building materials increases as the apparent density increases. The product of thermal conductivity and volumetric specific heat is a thermal property that has been shown to be important in its effect upon cooling load with unsteady flow of heat; this property increases very rapidly with an increase in apparent density. Empirical equations are offered that relate thermal conductivity and the product of thermal conductivity and volumetric specific heat to apparent density. These equations are approximate and should be used only when there are no reliable test data that give the desired properties.

Thermal Properties of Dry, Non-Metallic Solids (approximate):

Apparent densities, ρ, between 5 and 20 lb/ft³

Vegetable or mineral origin:

$$k = 0.0137\rho^{0.3} \text{ Btu/hr ft F}$$

Vegetable origin:

$$k\rho c = 0.0045\rho^{1.3} \text{ Btu}^2\text{/hr ft}^4 \text{ F}^2$$

Mineral origin:

$$k\rho c = 0.0027\rho^{1.3} \text{ Btu}^2\text{/hr ft}^4 \text{ F}^2$$

Apparent densities, ρ, between 20 and 60 lb/ft³

Vegetable or mineral origin:

$$k = 0.00118\rho^{1.12} \text{ Btu/hr ft F}$$

Vegetable origin:

$$k\rho c = 0.00039\rho^{2.12} \text{ Btu}^2\text{/hr ft}^4 \text{ F}^2$$

Mineral origin:

$$k\rho c = 0.00024\rho^{2.12} \text{ Btu}^2\text{/hr ft}^4 \text{ F}^2$$

Apparent densities, ρ, of 60 lb/ft³ and higher

Any origin:

$$k = \left(\frac{\rho}{143}\right)^{2.5} \text{ Btu/hr ft F}$$

Any origin:

$$k\rho c = \left(\frac{\rho}{55}\right)^{3.5} \text{ Btu}^2\text{/hr ft}^4 \text{ F}^2$$

The thermal properties of water are considerably out of line with those of non-metallic solids. The thermal conductivity of water is about 0.35 Btu/hr ft F at temperatures important in this study, while dry non-metallic solids with the same density as water will have thermal conductivities only one third of that of water. Similarly, $k\rho c$ for water is about 21.8 Btu²/hr ft⁴ F², while $k\rho c$ of any dry non-metallic solid of the same density as water is only about 7 per cent of that of water. It follows that the presence of moisture in materials of construction will considerably modify their thermal behaviors.

SOME CONCLUSIONS CONCERNING THERMAL BEHAVIOR OF COMPOSITE WALLS OR ROOFS

All conclusions summarized here apply only to the type of unsteady flow of heat defined in this study—periodic outdoor air temperature and incident solar radiation with constant temperature of the indoor air.

For any composite wall or roof of *sandwich* construction, there is an exact *thermal* equivalent in some simple, homogeneous material. Whether or not any available building material has the exact properties required of this equivalent material is another question. The significance of the first conclusion will be illustrated by examples. Consider a two layer wall with 12 in. of concrete on the outside and 1 in. of cork on the inside. The value of $(k\rho c)_s$ for an equivalent homogeneous wall may be shown to be 5.5, while $\left(\frac{L}{k}\right)_s = 4.3$. Plaster, or a non-metallic solid with an apparent density of 90 lb per cubic foot would have about this value of $k\rho c$; in a thickness of about 16 in., plaster or a non-metallic solid with an apparent density of 90 lb per cubic foot would have the same thermal resistance as the concrete and cork wall. Therefore, a simple, homogeneous wall of 16 in. of plaster is the *thermal* equivalent of the composite wall made up of 12 in. of concrete lined with 1 in. of cork; each wall would contribute approximately the same amount to the cooling load at the same time of day when exposed on the outside to a periodic air temperature and solar radiation while the temperature of the indoor air is held constant, provided the solar absorptivity of the exterior surface of each wall is the same.

As a second example, consider a composite wall made up as follows: 4 in. face brick, $^{25}/_{32}$ in. wood sheathing, air space, ½ in. insulating board and ½ in. plaster. The value of $(k\rho c)_s$ of the equivalent homogeneous wall may be found as 0.9, while the thermal resistance of the equivalent homogeneous wall must be 3.82. Pine would have substantially this value of $k\rho c$ and if used in a thickness of 3 in. would have about the same *thermal* resistance as the composite, brick veneer wall. Therefore, a simple, homogeneous wall of 3 in. of pine is the *thermal* equivalent of the composite brick veneer wall of this example and each wall would make approximately the same contribution to the cooling load at the same time of day provided the solar absorptivity of the exterior surface of each wall is the same.

A second conclusion is that the order of the materials in a composite wall or roof affects the thermal behavior of the construction. In the steady flow of heat, the order in which the successive layers of the materials are encountered in series has no effect upon the overall rate of heat conduction; this is not the case for periodic, unsteady flow of heat. A typical example of this conclusion is the case of a two layer composite construction consisting of 6 in. of concrete and 1 in. of rigid insulation. With the concrete on the outside, the time lag is about 5 hours; when the insulation is on the outside, the time lag is about 6 hours. In general, the facts concerning the thermal behavior of a composite wall of two materials in series are: if the two materials have the same solar absorptivities, interchanging the two materials will *not* affect the *daily* gain of heat, but use of the material of smaller density as the outdoor layer will *decrease* the *maximum* rate of heat transfer to the indoor air and *increase* the lag in time between the time of maximum rate of heat entry into

the outdoor layer and the time of maximum rate of heat transfer to the inside of the enclosure.

For a composite construction of three or more layers in series, this study indicates that interchanging any of the middle layers with the inside layer will have no great effect upon the thermal behavior of the wall or roof. Interchanging any of the middle layers or the inside layer with the *outside* layer, however, may affect the *maximum* rate of heat transfer to the indoor air and the time of that maximum. The direction of these changes is similar to the case of the two layer construction. If a material of low density (low thermal conductivity and low volumetric specific heat) in the middle or on the inside surface is exchanged with an outdoor layer of high density (high thermal conductivity and high volumetric specific heat), the result is a decrease in the maximum rate of heat transfer to the indoor air (if the solar absorptivities of the two materials are the same) and an increase in the time lag. In all cases, low solar absorptivity of the outside surface will *decrease* both the *daily* rate of heat transfer and the *maximum* rate of heat transfer. Time lag, however, is not influenced by a change in solar absorptivity, alone.

After finding the thermal resistance and the value of kpc of the simple, homogeneous wall or roof which is the thermal equivalent of the composite construction, the temperature of the inside surface and the contribution to the cooling load at any time of day may be found precisely as if the composite construction were a simple, homogeneous material. A complete summary of the work on periodic flow of heat through homogeneous and composite constructions will be given in a later paper.

APPENDIX

In general, the equations of Figs. A and B are for the purpose of determining two quantities; one of these is the harmonic decrement factor, λ, while the other is the harmonic lag angle, ϕ. Fig. A gives equations for a two-layer construction. Fig. B gives equations for a three-layer construction.

The primary variables upon which these quantities depend are all listed in the following definition of symbols. In general the subscript O is used to indicate the room-side or indoor layer of the material in the composite construction; the subscript m is used to indicate the middle layer; the subscript L is used to indicate the weather-side, or outdoor layer.

PRIMARY VARIABLES

Film Coefficients

h_O = inside air film coefficient of heat transfer, Btu/hr ft² F
h_L = outdoor air film coefficient of heat transfer, Btu/hr ft² F

Thermal Conductivities

k_O = thermal conductivity of material forming room-side layer, Btu/hr ft F
k_m = thermal conductivity of material forming middle-layer, Btu/hr ft F
k_L = thermal conductivity of material forming weather-side layer, Btu/hr ft F

Specific Heats

C_O = specific heat of material forming room-side layer, Btu/lb F
C_m = specific heat of material forming middle layer, Btu/lb F
C_L = specific heat of material forming weather-side layer, Btu/lb F

$$\lambda = \frac{2}{\sqrt{\left[r_b \frac{q_b}{q_c} + \pi_1 r_c \frac{q_c}{q_b}\right]^2 + \left[s_b \frac{q_b}{q_c} + \pi_1 s_c \frac{q_c}{q_b}\right]^2}}$$

$$\phi = \tan^{-1}\frac{\left[\frac{C_2}{\pi_3}+C_1\right]-\left[\frac{C_2}{\pi_3}+C_2\right]}{\left[\frac{C_2}{\pi_3}+C_1\right]+\left[\frac{C_2}{\pi_3}+C_2\right]} + \tan^{-1}\frac{\left[\frac{D_2}{\Gamma_3}+D_1\right]-\left[\frac{D_2}{\Gamma_3}+D_2\right]\left[s_b q_b^2+\pi_1 s_c q_c^2\right]-\left[\frac{D_2}{\Gamma_3}+D_1\right]+\left[\frac{D_2}{\Gamma_3}+D_2\right]\left[s_b q_b^2+\pi_1 s_c q_c^2\right]}{\left[\frac{D_2}{\Gamma_3}+D_1\right]+\left[\frac{D_2}{\Gamma_3}+D_2\right]\left[s_b q_b^2+\pi_1 s_c q_c^2\right]+\left[\frac{D_2}{\Gamma_3}+D_1\right]-\left[\frac{D_2}{\Gamma_3}+D_2\right]\left[s_b q_b^2+\pi_1 s_c q_c^2\right]}$$

where:

$$r_c = (C_1+2\pi_3 C_4)\left(\frac{C_2}{\pi_3}+C_1\right)+(C_2-2\pi_3 C_3)\left(\frac{C_2}{\pi_3}+C_2\right)$$

$$s_c = (C_2-2\pi_3 C_3)\left(\frac{C_2}{\pi_3}+C_1\right)-(C_1+2\pi_3 C_4)\left(\frac{C_2}{\pi_3}+C_2\right)$$

$$r_b = (D_1+2\Gamma_3 D_4)\left(\frac{D_2}{\Gamma_3}+D_1\right)+(D_2-2\Gamma_3 D_3)\left(\frac{D_2}{\Gamma_3}+D_2\right)$$

$$s_b = (D_2-2\Gamma_3 D_3)\left(\frac{D_2}{\Gamma_3}+D_1\right)-(D_1+2\Gamma_3 D_4)\left(\frac{D_2}{\Gamma_3}+D_2\right)$$

$$q_c = \sqrt{\frac{2}{\left[\frac{C_2}{\pi_3}+C_1\right]^2+\left[\frac{C_2}{\pi_3}+C_2\right]^2}}$$

$$q_b = \sqrt{\frac{2}{\left[\frac{D_2}{\Gamma_3}+D_1\right]^2+\left[\frac{D_2}{\Gamma_3}+D_2\right]^2}}$$

$C_1 = \cos\pi_2 \cosh\pi_2 + \sin\pi_2 \sinh\pi_2$ $\quad D_1 = \cos\Gamma_2 \cosh\Gamma_2 + \sin\Gamma_2 \sinh\Gamma_2$

$C_2 = \cos\pi_2 \cosh\pi_2 - \sin\pi_2 \sinh\pi_2$ $\quad D_2 = \cos\Gamma_2 \cosh\Gamma_2 - \sin\Gamma_2 \sinh\Gamma_2$

$C_3 = \sin\pi_2 \cosh\pi_2$ $\quad D_3 = \sin\Gamma_2 \cosh\Gamma_2$

$C_4 = \cos\pi_2 \sinh\pi_2$ $\quad D_4 = \cos\Gamma_2 \sinh\Gamma_2$

$$\pi_1 = \frac{h_o}{h_L} = 0.4125$$

$$\pi_2 = \sqrt{\frac{0.1309 A c_o}{k_o}}\, L_o \qquad \pi_3 = \frac{k_o}{h_o}\sqrt{\frac{0.1309 A c_o}{k_o}}$$

$$\Gamma_2 = \sqrt{\frac{0.1309 A c_L}{k_L}}\, L_L \qquad \Gamma_3 = \frac{k_L}{h_L}\sqrt{\frac{0.1309 A c_L}{k_L}}$$

FIG A.—EQUATIONS FOR DETERMINING λ AND ϕ FOR TWO-LAYER CONSTRUCTION

Densities

ρ_O = density of material forming room-side layer, lb/ft³
ρ_m = density of material forming middle layer, lb/ft³
ρ_L = density of material forming weather-side layer, lb/ft³

Thicknesses

L_O = thickness of material forming room-side layer, ft
L_m = thickness of material forming middle layer, ft
L_L = thickness of material forming weather-side layer, ft

The remaining symbols represent functions introduced in the mathematical derivations. These need no definition other than the defining equations which appear in Figs. A and B.

$$\lambda = \eta \cdot \delta \cdot \gamma \qquad \phi = \upsilon + \epsilon + \mu$$

$$\eta = \sqrt{\frac{2}{\left[\frac{C_3}{\pi_3}+C_1\right]^2+\left[\frac{C_4}{\pi_3}+C_2\right]^2}} \qquad \upsilon = \tan^{-1}\frac{\left[\frac{C_3}{\pi_3}+C_1\right]-\left[\frac{C_4}{\pi_3}+C_2\right]}{\left[\frac{C_3}{\pi_3}+C_1\right]+\left[\frac{C_4}{\pi_3}+C_2\right]}$$

$$\delta = \frac{P_w}{\sqrt{\left[\frac{y_w P_w^2}{2}+\frac{1}{\Omega_3}\frac{r_c q_c^2}{2}\right]^2+\left[\frac{z_w P_w^2}{2}+\frac{1}{\Omega_3}\frac{s_c q_c^2}{2}\right]^2}}$$

$$\epsilon = \tan^{-1}\frac{(w_3-w_4)\left[\frac{y_w P_w^2}{2}+\frac{1}{\Omega_3}\frac{r_c q_c^2}{2}\right]-(w_3+w_4)\left[\frac{z_w P_w^2}{2}+\frac{1}{\Omega_3}\frac{s_c q_c^2}{2}\right]}{(w_3+w_4)\left[\frac{y_w P_w^2}{2}+\frac{1}{\Omega_3}\frac{r_c q_c^2}{2}\right]+(w_3-w_4)\left[\frac{z_w P_w^2}{2}+\frac{1}{\Omega_3}\frac{s_c q_c^2}{2}\right]}$$

$$\gamma = \frac{q_0}{\sqrt{\left\{\frac{r_0 q_0^2}{2}-\pi_1\Omega_3\frac{P_w^2}{2}\left[(w_3+w_4)\delta\cos\epsilon-(w_3-w_4)\delta\sin\epsilon-y_w\right]\right\}^2+\left\{\frac{s_0 q_0^2}{2}-\pi_1\Omega_3\frac{P_w^2}{2}\left[(w_3-w_4)\delta\cos\epsilon-(w_3+w_4)\delta\sin\epsilon-z_w\right]\right\}^2}}$$

$$\mu = \tan^{-1}\frac{\left[\left(\frac{D_2}{r_3}+D_1\right)-\left(\frac{D_4}{r_3}+D_3\right)\right]\left\{\frac{r_0 q_0^2}{2}-\pi_1\Omega_3\frac{P_w^2}{2}\left[(w_3+w_4)\delta\cos\epsilon-(w_3-w_4)\delta\sin\epsilon-y_w\right]\right\}}{\left[\left(\frac{D_2}{r_3}+D_1\right)+\left(\frac{D_4}{r_3}+D_3\right)\right]\left\{\frac{r_0 q_0^2}{2}-\pi_1\Omega_3\frac{P_w^2}{2}\left[(w_3+w_4)\delta\cos\epsilon-(w_3-w_4)\delta\sin\epsilon-y_w\right]\right\}}$$

$$\sqrt{\frac{-\left[\left(\frac{D_2}{r_3}+D_1\right)+\left(\frac{D_4}{r_3}+D_3\right)\right]\left\{\frac{s_0 q_0^2}{2}-\pi_1\Omega_3\frac{P_w^2}{2}\left[(w_3-w_4)\delta\cos\epsilon-(w_3+w_4)\delta\sin\epsilon-z_w\right]\right\}}{+\left[\left(\frac{D_2}{r_3}+D_1\right)-\left(\frac{D_4}{r_3}+D_3\right)\right]\left\{\frac{s_0 q_0^2}{2}-\pi_1\Omega_3\frac{P_w^2}{2}\left[(w_3-w_4)\delta\cos\epsilon-(w_3+w_4)\delta\sin\epsilon-z_w\right]\right\}}}$$

where: $y_w = w_1 w_3 + w_2 w_4$ $\qquad$ $w_1 = \cos\Omega_2\cosh\Omega_2 + \sin\Omega_2\sinh\Omega_2$

$z_w = w_2 w_3 - w_1 w_4$ $\qquad$ $w_2 = \cos\Omega_2\cosh\Omega_2 - \sin\Omega_2\sinh\Omega_2$

$P_w = \sqrt{\frac{2}{w_3^2+w_4^2}}$ $\qquad$ $w_3 = \sin\Omega_2\cosh\Omega_2$

$w_4 = \cos\Omega_2\sinh\Omega_2$

$$\Omega_2 = \sqrt{\frac{0.1309\rho_m c_m}{k_m}}\,L_m \qquad \Omega_3 = \frac{k_m}{k_0}\sqrt{\frac{0.1309\rho_m c_m}{k_m}}$$

and the following functions are defined in Fig. A

$\pi_1, \pi_2, \pi_3, C_1, C_2, C_3, C_4, q_c, r_c, s_c$

$r_2, r_3, D_1, D_2, D_3, D_4, q_0, r_0, s_0$

FIG. B.—EQUATIONS FOR DETERMINING λ AND ϕ FOR THREE-LAYER CONSTRUCTION

TABLE A—SAMPLE RESULTS FOR TWO-LAYER WALLS

WALL NO.	PROPERTIES OF INSIDE MATERIAL		PROPERTIES OF OUTSIDE MATERIAL		FUNDAMENTAL VALUES CALCULATED FROM PRECISE EQUATIONS	
	$\left(\frac{L}{k}\right)_0$	$(k\rho c)_0$	$\left(\frac{L}{k}\right)_L$	$(k\rho c)_L$	λ	Time Lag, hr
1	0.606	5.20	1.250	4.89	0.1613	4.52
2	1.212	5.20	1.250	4.89	0.0877	6.15
3	2.424	5.20	1.250	4.89	0.0359	10.31
4	3.636	5.20	1.250	4.89	0.0132	14.2
5	1.212	5.20	0.625	4.89	0.1613	4.55
6	1.212	5.20	2.500	4.89	0.0361	10.3
7	1.212	5.20	3.750	4.89	0.0133	14.2
8	1.212	5.20	5.000	4.89	0.0049	17.9
9	0.606	5.20	0.500	30.6	0.1776	5.37
10	1.212	5.20	0.500	30.6	0.1088	7.09
11	3.636	5.20	0.500	30.6	0.0150	14.8
12	1.212	5.20	0.250	30.6	0.1810	4.92
13	1.212	5.20	2.500	30.6	0.0019	23.0
14	0.606	5.20	2.500	1.223	0.1178	3.88
15	1.212	5.20	2.500	1.223	0.0717	6.30
16	3.636	5.20	2.500	1.223	0.0094	14.0
17	1.212	5.20	0.500	1.223	0.1858	3.56
18	1.212	5.20	1.250	1.223	0.1240	4.62
19	1.212	5.20	0.250	1.223	0.2184	3.13
20	2.424	0.052	1.250	5.20	0.1017	3.66
21	1.212	4.89	2.500	0.049	0.0851	4.38
22	1.212	20.8	0.500	30.6	0.0439	11.6
23	1.212	0.208	0.500	30.6	0.1399	4.63
24	1.212	83.2	0.500	30.6	0.0049	19.6
25	1.212	5.20	0.500	0.306	0.1882	3.4
26	1.212	20.8	0.500	122.3	0.0147	15.8
27	1.212	83.2	0.500	122.3	0.0019	24
28	1.212	20.8	0.500	1.22	0.0658	8.47
29	3.636	5.20	0.250	30.6	0.0249	12.6

TABLE B—SAMPLE RESULTS FOR THREE-LAYER WALLS

WALL NO.	PROPERTIES OF INSIDE MATERIAL		PROPERTIES OF MIDDLE MATERIAL		PROPERTIES OF OUTSIDE MATERIAL		FUNDAMENTAL VALUES CALCULATED FROM PRECISE EQUATIONS	
	$\left(\frac{L}{k}\right)_0$	$(k\rho c)_0$	$\left(\frac{L}{k}\right)_m$	$(k\rho c)_m$	$\left(\frac{L}{k}\right)_L$	$(k\rho c)_L$	λ	Time Lag, hr
31	0.606	5.20	0.606	5.20	0.250	4.89	0.2142	3.38
32	2.424	5.20	1.212	5.20	0.625	4.89	0.0219	12.2
33	0.606	5.20	0.606	5.20	0.625	4.89	0.1612	4.6
34	0.606	5.20	0.606	5.20	0.500	1.22	0.1858	3.6
35	0.606	5.20	0.606	5.20	0.500	30.6	0.1088	7.1
36	0.606	5.20	0.606	5.20	1.25	1.22	0.1240	4.6
37	0.606	5.20	0.606	5.20	1.25	4.89	0.0959	6.5
38	0.606	5.20	0.606	5.20	0.250	1.22	0.2184	3.1
39	0.606	5.20	2.424	5.20	0.250	4.89	0.0489	9.2
40	2.424	20.8	1.212	0.208	0.625	4.89	0.0039	17.9
41	1.212	0.208	2.424	20.8	0.625	4.89	0.0037	17.1
42	2.424	5.20	1.212	0.208	0.500	1.22	0.0386	8.4
43	1.212	0.208	2.424	5.20	0.500	1.22	0.0384	8.5

HARMONICS

The equations given in Fig. A (two-layer construction) for the functions π_2, π_3, Γ_2, and Γ_3 are for the fundamental (or first harmonic) values. When these values are substituted in the equations for λ and ϕ, fundamental values of these quantities are found. For the harmonics of order η, find new values of π_2, π_3, Γ_2, and Γ_3 by multiplying the fundamental values by $\sqrt{\eta}$. These new values should be used in finding the harmonic values of λ and ϕ.

Similarly, the equations given in Fig. B (three-layer construction) for Ω_2 and Ω_3 are for the fundamental values. For the harmonics of order η, find new values of Ω_2 and Ω_3 by multiplying the fundamental values by $\sqrt{\eta}$.

TWO-LAYER WALLS OR ROOFS

It is assumed that:

1. The temperature of the outdoor air and the total solar radiation incident upon the outer surface of the wall or roof are cyclic with a period of 24 hours.
2. The temperature of the indoor air is constant.
3. The rate of heat transfer from the outdoor air to the surface is constant and equal to 4 Btu/hr ft² F.
4. The rate of heat transfer from the indoor surface is constant and equal to 1.65 Btu/hr ft² for each degree of temperature difference between the surface and indoor air.
5. The wall or roof is composed of two layers arranged in series in the direction of heat flow, each layer being itself a single, homogeneous material.
6. The two layers make perfect thermal contact at their adjoining faces.

The equations for the decrement factor λ and the lag angle ϕ for the two-layer wall or roof are given in Fig. A.

In Table A are given fundamental decrement factors and time lags for 29 walls as calculated from the equations of Fig. A.

THREE-LAYER WALLS OR ROOFS

The assumptions made for the three-layer case are the same as those for the two-layer case with the single exception of the number of layers involved. The equations for the decrement factor and lag angle for the three-layer wall or roof are given in Fig. B.

In Table B are given fundamental decrement factors and time lags for 13 sample walls as calculated from the equations of Fig. B.

DISCUSSION

CYRIL TASKER, Cleveland, Ohio (WRITTEN): I think that it would be quite wrong for us to let this opportunity pass to pay tribute not only to Professors Mackey and Wright for the work that they have done for so long in cooperation with the Society, but to the man who by his energy and persistence has definitely spurred on these studies. If every Technical Advisory Committee was as active as the one on Cooling Load, the Laboratory staff would not begin to be large enough. Mr. Zieber has inspired the members of his Committee with the importance of the studies carried on under their direction and has kept in close touch with Professor Mackey throughout all the work at Cornell.

As a result of his energetic direction there are a large number of projects under way at the Laboratory and at the cooperating institutions which are directly or indirectly part of the whole problem of cooling load and periodic heat flow.

Professor Mackey has indicated the need for certain types of data which the Weather Bureau might provide. It is only by our proving that these data are valuable and important to the industry that we can obtain the cooperation of the Weather Bureau in providing them. Our ultimate aim is, of course, to get the data into such form as will allow their being presented in relatively simple language and tables in THE GUIDE. The Committee on Research is taking steps to have all available data correlated so that the Guide Chapter on Cooling Load will be a Chapter that will find ready use by the membership and save engineers from a great number of complex calculations when figuring cooling loads.

C. M. ASHLEY, Syracuse, N. Y. (WRITTEN): This series of papers have been a most valuable contribution to the Society. One of their primary contributions has been to show the relatively small importance of the heat load of buildings due to sunlight radiation on walls. It has contributed greatly in putting our understanding of the periodic heat flow problem on the basis of consistent and orderly procedure, and has helped to show the many weaknesses and inconsistencies involved in our experimental data. These programs have demonstrated clearly the need for better supplementary data on conductivity, density, and specific heat of materials, film coefficients, sunlight radiation, etc., and have stimulated research activity on these subjects.

I feel that the data is still somewhat unwieldy for use by the estimating engineer and it is greatly to be hoped that the authors can suggest practical means by which the data can be reduced to a form which will be more readily usable.

AUTHOR'S CLOSURE: A simplified presentation of the data has been given to the TAC on Cooling Load and it is hoped that simple tables of desired accuracy can be provided. Several additional points, not covered in the abridged paper, require further discussion. First, it should be pointed out that the exact analytical solutions for the two- and three-layer walls are included in the Appendix.

The effect of material order was not illustrated by examples in the paper. A pair of examples will serve to show the effect of interchanging inside and outside layers. The walls considered are composed of concrete ($k = 1$ Btu/hr ft F; $kpc = 35$) and corkboard ($k = 0.025$ Btu/hr ft F; $kpc = 0.10$). Table C gives the fundamental decrement factor and time lags calculated from the accurate equations for four walls. The materials chosen for these examples have as widely different thermal properties as will normally be found in practice. This wide difference in thermal properties accentuates the effects of the reversal of materials in a wall. As shown in Table C, it will be noticed in each case that the wall having the material of lower density on the outside gives the larger time lag and the smaller decrement factor. Also, the reversal affects the time and rate of maximum heat transfer more in the case of wall with a small time lag. *For most practical building walls or roofs the effect of order of the materials is small enough that it may well be neglected in cooling load calculations.*

Equation 2 given in the paper for the property $(kpc)_e$ for the single layer wall equivalent to a multi-layer wall is somewhat limited in application. For example, in the case of a two-layer wall with the thermal resistance of the inside layer more than

TABLE C—FUNDAMENTAL FACTOR AND TIME LAG

OUTSIDE LAYER	INSIDE LAYER	FUNDAMENTAL DECREMENT FACTOR	FUNDAMENTAL TIME LAG, HOURS
2 in. concrete	1 in. corkboard	0.126	1.95
1 in. corkboard	2 in. concrete	0.107	2.98
12 in. concrete	1 in. corkboard	0.0209	9.45
1 in. corkboard	12 in. concrete	0.0145	10

TABLE D—FUNDAMENTAL TIME LAGS AND DECREMENT FACTORS FOR CONCRETE AND CORKBOARD WALLS (OR ROOFS)

Thickness of Outside Concrete, Inches	Thickness of Inside Corkboard, Inches	A Time Lag for Simple Concrete Wall, Hrs.	B Time Lag for Simple Corkboard Wall, Hrs.	A + B Approx. Time Lag for Composite Wall, Hrs.	Correct Time Lag for Composite Wall, Hrs.	$\left(\frac{L}{k}\right)_a$	$(kpc)_a$ Based on $\left(\frac{L}{k}\right)_a$ and Approx. Time Lag	Approx. Fundamental Decrement Factor	Correct Fundamental Decrement Factor
2	1	1.3	0.3	1.6	1.95	3.50	0.60	0.130	0.126
	2		0.9	2.2	2.80	6.83	0.25	0.072	0.075
	3		2.0	3.3	3.66	10.17	0.18	0.046	0.047
	4		3.2	4.5	4.60	13.50	0.15	0.031	0.031
	6		6.0	7.3	7.75	20.17	0.13	0.015	0.016
4	1	2.8	0.3	3.1	3.6	3.67	1.0	0.110	0.094
	2		0.9	3.7	4.7	7.00	0.40	0.055	0.056
	3		2.0	4.8	5.5	10.33	0.28	0.037	0.035
	4		3.2	6.0	6.4	13.67	0.20	0.026	0.023
	6		6.0	8.8	9.7	20.33	0.16	0.011	0.011
6	1	4.2	0.3	4.5	5.2	3.83	1.5	0.092	0.066
	2		0.9	5.1	6.0	7.17	0.60	0.045	0.038
	3		2.0	6.2	7.0	10.50	0.36	0.032	0.024
	4		3.2	7.4	8.1	13.83	0.28	0.014	0.016
	6		6.0	10.2	11.2	20.50	0.20	0.009	0.0081
8	1	5.7	0.3	6.0	6.4	4.00	2.0	0.071	0.044
	2		0.9	6.6	7.5	7.33	0.80	0.036	0.026
	3		2.0	7.7	8.5	10.67	0.50	0.024	0.016
	4		3.2	8.9	9.6	14.00	0.32	0.017	0.011
	6		6.0	11.7	12.6	20.67	0.25	0.0063	0.0055
10	1	7.0	0.3	7.3	7.8	4.17	2.5	0.055	0.030
	2		0.9	7.9	8.9	7.50	0.95	0.032	0.018
	3		2.0	9.0	9.9	10.83	0.60	0.020	0.011
	4		3.2	10.2	11.0	14.17	0.42	0.012	0.0074
	6		6.0	13.0	14.0	20.83	0.30	0.0046	0.0038
12	1	8.5	0.3	8.8	9.4	4.33	3.0	0.040	0.021
	2		0.9	9.4	10.3	7.67	1.2	0.022	0.012
	3		2.0	10.5	11.2	11.00	0.75	0.016	0.0076
	4		3.2	11.7	12.3	14.33	0.50	0.010	0.0051
	6		6.0	14.5	15.3	21.00	0.35	0.0038	0.0026

ten times the thermal resistance of the outside layer, the second numerator term becomes negative and an incorrect value of the property $(kpc)_a$ is obtained. Another method of obtaining an equivalent simple wall was developed after the preparation of the paper. This newer method, which does not take material order into account, is given in 5 steps as follows:

1. Find the time lag for each material as if it were a simple wall or roof. (See Fig. 2 of a previous paper.[2]

[2] RESEARCH REPORT No. 1255—Periodic Heat Flow—Homogeneous Walls or Roofs, Mackey and Wright. (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 293.)

2. Add the time lags thus obtained to find the (approximate) time lag of the composite wall or roof.

3. Find the thermal resistance of the equivalent simple wall or roof by adding the resistances of the materials in the composite wall or roof.

4. Find the product $(k\rho c)_s$ for the equivalent simple wall or roof from the thermal resistance obtained in Step 3 and the time lag obtained in Step 2. (See Fig. 2 of a previous paper.[3]) On this figure, use time lag from Step 2 as ordinate and conductance $\left(\frac{K}{L}\right)_s$ of equivalent simple wall or roof as abscissa; by interpolation between lines of constant $k\rho c$, read $(k\rho c)_s$.

5. Find the fundamental decrement factor from the thermal resistance obtained in Step 3 and the product $(k\rho c)_s$ found in Step 4. (See Fig. 1 of a previous paper.[3])

Table D gives the correct fundamental time lags and decrement factors for a series of concrete and corkboard walls (or roofs) and the results of the approximate method just outlined. The correct and approximate time lags differ at most by 1½ hours. The difference between the correct and approximate decrement factors may appear large at first sight, but in terms of rates of heat transfer to the indoor air the differences are minor.

[3] Op. cit. Note 2.

1300

No. 1300

HEAT LOSSES THROUGH WETTED WALLS

By E. C. WILLEY,* CORVALLIS, ORE.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station, Oregon State College.

FOR many years heating engineers of the northwest country have conceded that walls exposed to the wind and rain have a much higher heat loss than normally would be calculated from their wall coefficients and the degree day records.

The late F. C. Houghten, director of the A.S.H.V.E. Research Laboratory, suggested that studies be made to determine the effect of wetted surface on the overall heat loss of a wall by placing heat flow meters on walls which were exposed to the prevalent rains and winds. An attempt was made to do this, but it so happened that the following two winters were so mild and dry that it was impossible to get a sufficient number of significant readings. It was decided, therefore, that the quickest and most reliable way to obtain this information would be to build equipment to reproduce the conditions mechanically.

Late in 1940 a committee of members of the Oregon chapter of A.S.H.V.E. met to study the possibilities of building suitable equipment and carrying on this research project. This committee consisted of J. D. Kroeker, *chairman*, T. H. McClung, J. F. McIndoe, J. E. Yates and E. C. Willey. Subsequent members were T. E. Taylor, W. J. Kollas, W. B. Morrison, Albert Freeman and D. E. Cooper.

This committee first enlisted the assistance of manufacturers and jobbers in and around Portland, Ore., and obtained promises of support of the project in the form of materials and supplies. With these promises and the cooperation of the Oregon State College Engineering Experiment Station, the building of the equipment was started. The work progressed slowly because of war conditions, especially the scarcity of materials and labor.

In the fall of 1942, the Committee on Research made funds available for the continuation of the project and, though the work progressed somewhat

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faster, many unforeseen difficulties arose, some requiring rebuilding important parts of the equipment.

EQUIPMENT

The general arrangement of test equipment (see Fig. 1) was similar to the hot box cold room used by the University of Minnesota.[1] This consisted of a cold room with floor area 10 ft x 10 ft and ceiling 8 ft. In the center of one

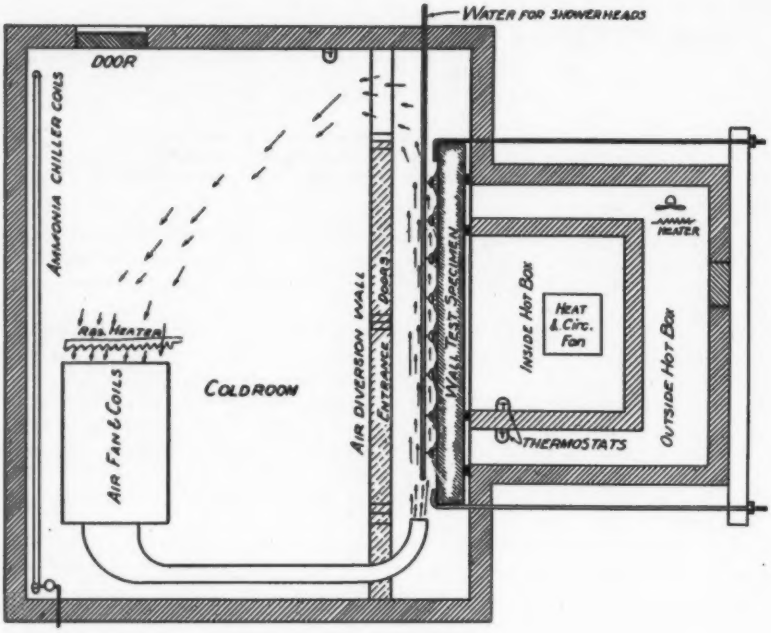


FIG. 1. PLAN OF COLD ROOM

wall there was an opening 5 ft x 5 ft over which (on the outside) a hot box was built with dimensions of 5 ft x 5 ft x 5 ft. This box was termed the outer hot box. Suspended in the center of the outer hot box was an inner hot box 3 ft x 3 ft x 3 ft. The inner edges of the outer and inner hot boxes were flush and padded with heavy felt against which the test wall specimen was secured.

A fan located in the cold room delivered air through a duct to the surface of the test wall. The exhaust opening of the duct was 6 in. x 6 ft and was

[1] Thermal Conductivity of Building Materials, *Bulletin No. 12*, Engineering Experiment Station, University of Minnesota.

arranged so that the air was distributed evenly over the surface of the wall specimen.

Near the top of the wall specimen a series of small water sprays was arranged to wet the wall and simulate rain conditions. Water supply for this was taken either directly from the city water supply or precooled to any desired temperature. Refrigeration for the cold room was supplied through an ammonia compressor and direct expansion coils in the room.

Temperatures were measured by copper constantan thermocouples and a potentiometer. The thermocouples were placed four on the outside and four on the inside of the test wall specimen. These were located 6 in. out from the wall and equidistant from the center so that each thermocouple would

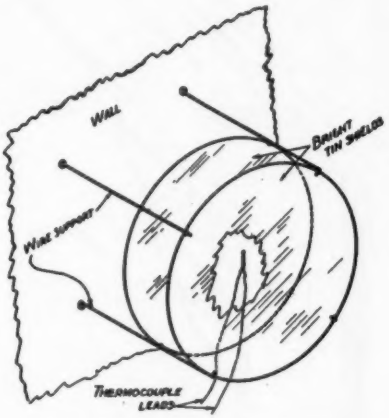


FIG. 2. THERMOCOUPLE SHIELD

register the temperature of about one-fourth of the wall section. Thermocouples were also placed on the surface at center of the test wall. In addition, there were couples placed in the center (outside and inside) of each of the walls of the inner hot box to register the temperature of the inside and outside hot boxes and assure that no heat left the inner hot box except through the test portion of the wall. All thermocouples were shielded, as shown in Fig. 2, to avoid radiation effect and all were connected by leads to a multipoint switch and through a common cold junction to a potentiometer.

The test wall (Fig. 3) was of frame construction consisting of wood lath and ½ in. plaster on the inside; 2 x 4 studding, ¾ in. shiplap, tar paper and 6 in. cedar siding on the outside. The siding was given three coats of outside white paint.

Heat was supplied to the inside hot box by means of electric resistance heaters. These were enclosed in a box so that no part of the heater was directly exposed to the test wall or the sides of the inside hot box. A small

electric fan provided air circulation in the heater box, bringing in the air at the bottom and distributing it in all directions from the top. (See Fig. 4.)

The outside hot box also was warmed by means of an electric heater which in turn was shielded from direct radiation to the inside hot box and air in this case too was distributed by means of an electric fan.

Temperatures of the inside and outside hot boxes were controlled by thermostats through magnetic switches to the power sources. Electric power measurement was made with a watt hour meter which was read to one tenth

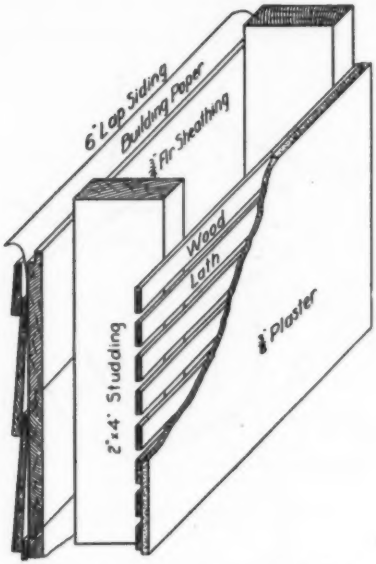


FIG. 3. CONSTRUCTION OF TEST WALL

watt hour. The watt hour meter was checked by a standard test meter on each reading.

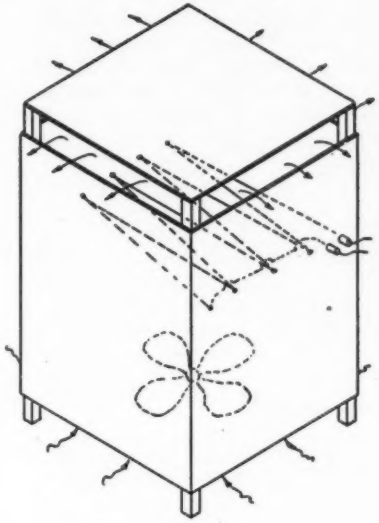
The walls of the cold room were constructed of 2 in. x 4 in. studding with fir plywood on each side. Space between the studding was filled with cedar shavings. The ceiling had plywood on the inside and the joists were filled with 6 in. of cedar shavings. The outside hot box was similar to the cold box in construction. The inside hot box was constructed of three layers of 1 in. insulating board encased on each side with fir plywood (see Fig. 5).

As shown in Fig. 1, the plan view of the test apparatus, the test wall is held against the hot boxes by means of long screw clamps and is sealed by heavy felt padding. The diversion wall aids in directing the air streams across the face of the test wall and also prevents the moisture from the shower heads

getting into the cold room. This diversion wall is removable in order to permit changing the test walls.

A large metal trough was installed under the test wall to collect the water, which could be recirculated or diverted to the sewer. A water recirculation pump was provided to circulate the water through a cooling coil in the fan system to maintain any desired temperature.

Thermostatic controls actuated the expansion valve in the ammonia refrigeration line to control the cold room temperatures. Also another control



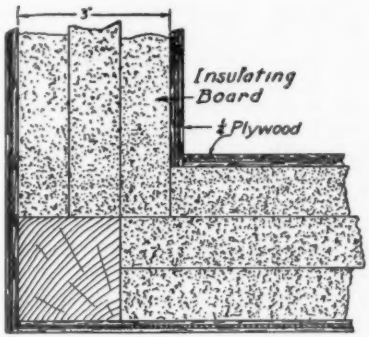


FIG. 5. WALL CONSTRUCTION OF INSIDE HOT BOX

FIG. 4. INSIDE HOT BOX AND ELECTRIC RESISTANCE HEATERS

consisted of an electric resistance heater placed at the intake of the circulation fan which would act if the temperatures dropped below any set minimum temperature.

TEST PROCEDURE

In general the tests were made to determine the additional heating load which would result from the wetting of the outer surface of a wall. As this was a relative condition, it was not considered necessary to make tests on many walls but only to determine the relative results on one type of wall. The wall used was of a type which is very common in the Pacific Northwest country. The first group of tests was made to determine the overall heat transfer coefficient U for the wall under conditions of still air. Temperatures of 72 F to 74 F were used throughout all the series for the inside temperature. The cold room temperatures varied from 30 F to 45 F.

The second group of tests was similar to the first but was made with air movements of from ten to twenty miles per hour. In both the first and second

group, the coefficients obtained checked very closely to those given in the HEATING, VENTILATING, AIR CONDITIONING GUIDE for this specific construction of wall.

The third group of tests was made using the water spray. This group was divided into two parts. The first was made by varying the water temperatures from 36 F to 50 F and keeping the amount of water flow constant. The second was made using constant water temperature but varying the flow from an equivalent of ¼ in. rainfall to ¾ in. rainfall per hour. In each of these tests, calculations were made to determine the general overall coefficient U with the specific conditions.

All tests were preceded by a long warming up period to insure a condition of equilibrium. Then readings were taken every 15 min of the following: (1) temperatures of cold room, four points on the cold room side of wall and on the center surface of wall, four points on the inside of test wall, five wall temperatures on inside of inside hot box and the five temperatures on the outside wall of inside hot box; (2) temperature of spray water (when spray was used); (3) reading of the watt meters.

RESULTS

The general results of this investigation may be summed up in a few words. However, of equal importance are some of the findings in the early part of the investigation which resulted in changes in the equipment and procedure.

When the equipment was first ready for tests, there were no provisions for guarding the thermocouples. The result was that temperatures varied much

SUMMARY OF TEST RESULTS

DATE OF RUN	CONDITION	$t_2 - t_1$* F DEG	DURATION HOURS	TOTAL WATT HR	COEF. U BTU/HR/SQ FT/ F DEG
8– 2–45	Dry, still air	33.0	6	126	0.241
8–27–45		32.9	6	126	0.241
9– 4–45		31.6	6	121	0.242
12– 5–45		41.3	6	157	0.240
					0.241 avg.
8–31–45	Dry, 15 mile wind	33.2	12	312	0.296
12– 6–45		41.2	5	151	0.278
12– 9–45		44.6	2	65	0.276
12–11–45		46.1	28	963	0.283
					0.283 avg.
8–25–45	15 mile wind	33.2	11	307	0.316
9– 1–45	¼ in. rainfall	32.4	16	439	0.322
9– 5–45	sweeping outside	32.1	7	190	0.321
11–24–45		35.0	1	29	0.314
12– 8–45		40.3	5.5	186	0.318
					0.318 avg.

* The average temperature of all runs on the hot side was 72 F.

more than seemed reasonable. Also the heaters were exposed to view of the walls and the temperatures were found to be much higher on the walls nearest the heater. This was rectified by shielding the couples. It was still noticed that when the fan was running, the temperatures would level off; but when the fan stopped, they would immediately begin changing. It was, therefore, decided to allow the fans to run constantly during the tests.

The question as to the significance of temperature of the rain arose soon after the first few tests were made using the wet surface.

Investigation at the offices of the weather bureau showed that no records had been kept of the actual temperature of the rain, but it was very evident

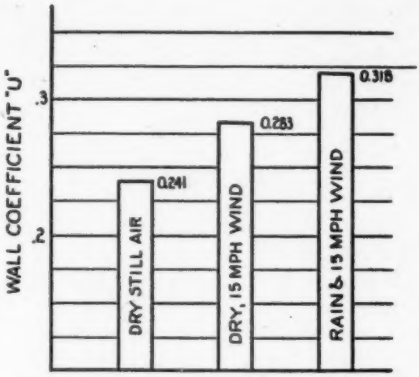


FIG. 6. COMPARISON OF AVERAGE OVERALL HEAT TRANSFER COEFFICIENTS FOR THE THREE CONDITIONS OF TEST

that it was not always the same temperature as the surrounding air since even in warm weather hail may fall at 32 F.

Rather than spending a winter of investigation to determine the average temperature of the rainfall, a series of tests was made in the test apparatus using different temperatures of water on the wall. The results of these were very gratifying as the heat loss, in all cases using water from 36 F to 50 F which was both colder and warmer than the surrounding air, remained practically constant. This indicated that the evaporation of the moisture rather than the temperature was causing the cooling effect.

The final average overall coefficient U from all tests with the wall under dry still air conditions was 0.241 Btu per (hr) (sq ft) (Fahrenheit degree). The final average of all tests with the dry wall exposed to an average 15 mph wind was 0.283. The final average of tests conducted under wind and rain conditions was $U = 0.318$ (see Fig. 6).

CONCLUSIONS

From the results of these experiments, it is very evident that rain conditions will increase the heat loss through a wall. This may not be of sufficient

proportion to warrant adding radiation, but should be given consideration in calculating the yearly fuel bills.

The bar chart, Fig. 6, shows that the wetted wall surface will add as much heat load to a wall as the normal 15 mph wind adds to the still air condition. The difference between $U = 0.283$ for the dry wind condition to $U = 0.318$ in the wet condition is 12.4 per cent.

When calculating the heat loss of a building for the purpose of estimating fuel consumption, consideration should be given to those walls which are exposed to the prevailing rains and wind. To these an additional load of 12.4 per cent should be added for that part of the season when the rain and winds prevail. This would not be at a time when the heat loss was maximum, of course, for at that time the weather would be freezing.

It has long been a common remark among building owners after a mild but rainy season, *even though we have had a mild winter, I have used more fuel than last winter when it was really cold.* This, no doubt, is due to a great extent to the 12.4 per cent additional load caused by the wet surface.

DISCUSSION

W. H. CARRIER, Syracuse, N. Y.: Was any attempt made to apply a correction for different outdoor vapor pressures?

C. F. KAYAN, New York, N. Y.: Had the outside film coefficients been determined for the test conditions?

CYRIL TASKER, Cleveland, Ohio: I know of some of the difficulties which Professor Willey experienced in carrying this work to completion. Much of the work was done personally in spare time, weekends and holidays, and he deserves credit and our thanks for his persistence in carrying it through.

PROFESSOR WILLEY: No work was done in determining film coefficients as the investigation covered only the determination of overall coefficients.

1301

No. 1301

A RATIONAL BASIS FOR SOLAR HEATING ANALYSIS †

By F. W. HUTCHINSON * AND W. P. CHAPMAN,** LAFAYETTE, IND.

INTRODUCTION

THE determination of gains or losses resulting from the use of window area in so called solar construction is complicated by many factors; not least of which is the difficulty of effectively integrating, diurnally and seasonally, the varying quantities of solar energy that are transmitted through or absorbed by the window. The present paper develops a rational general equation which can be used to obtain the gain or loss, in Btu per heating season, resulting from the use of unit area of window in place of an equal area of non-transmitting wall. This equation which is developed as Equation 23*a* in this paper is,

$$Q = F \sum_{n=1}^{n=5088} \nabla_n - 5088(U_c - U_w)(t_i - t_o')$$

where,

Q = annual saving (based on October 1 to May 1 heating season) in Btu due to replacing one square foot of non-transmitting *south* wall with a double glass *solar* window which is 100 per cent in the shade (due to roof overhang) *only* at solar noon on June 21 and 100 per cent irradiated by noon sun (solar time) *only* on January 21.

t_i = the design inside air temperature.

t_o' = the *normal* outside air temperature during the seven month heating season from October 1 to May 1 (Values given on pp. 258–9 of the HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1946.)

U_w = overall coefficient of heat transfer of the unit area of wall which is to be replaced by unit area of window.

U_c = overall coefficient of heat transfer of window, including inside and outside air films, two sheets of identical glass and separating air space.

5088 = number of hours in the selected seven month heating season.

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† An analytical paper prepared in cooperation with the Housing Research Division, Purdue Research Foundation; G. Stanley Meikle, Research Director of the Foundation; Carl F. Boester, Housing Executive.

* Professor of Mechanical Engineering, Purdue University. Member of A.S.H.V.E.

** Research Fellow, Purdue University.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Montreal, Que., Canada, June, 1946.

F = fraction of maximum possible sunshine which occurs during an average year in the 7 month heating season (refer to Table 1 for values in representative cities).

$\sum_{n=1}^{n=5088} \nabla_n$ = the summation of solar term, ∇_n, for each of the 5088 hours of the heating season.

$$\nabla_n = E\left\{ U_c\left[(r_o + 1.5r_g + r_a)q_{a2} + \left(\frac{r_d}{2} + r_o\right) q_{a1}\right] + q_i\right\}$$

TABLE 1—VALUES OF USAGE RATIO, F, FOR REPRESENTATIVE CITIES

[F is ratio of the average number of sunshine hours in the period from October 1 to May 1 to the maximum possible sunshine hours (at the latitude of the city[a]) for the same period.]

CITY	F	CITY	F
Albany, N. Y.	0.463	Jacksonville, Fla.	0.400
Albuquerque, N. M.	0.770	Joliet, Ill.	0.530
Atlanta, Ga.	0.522	Lincoln, Neb.	0.614
Baltimore, Md.	0.553	Little Rock, Ark.	0.513
Birmingham, Ala.	0.510	Louisville, Ky.	0.514
Bismarck, N. D.	0.546	Madison, Wis.	0.504
Boise, Ida.	0.540	Minneapolis, Minn.	0.527
Boston, Mass.	0.540	Newark, N. J.	0.550
Burlington, Vt.	0.419	New Orleans, La.	0.370
Chattanooga, Tenn.	0.503	Phoenix, Ariz.	0.590
Cheyenne, Wyo.	0.666	Portland, Me.	0.525
Cleveland, O.	0.408	Providence, R. I.	0.542
Columbia, S. C.	0.511	Raleigh, N. C.	0.570
Concord, N. H.	0.515	Reno, Nev.	0.637
Dallas, Tex.	0.470	Richmond, Va.	0.594
Davenport, Ia.	0.539	St. Louis, Mo.	0.567
Denver, Colo.	0.705	Salt Lake City, Utah	0.592
Detroit, Mich.	0.429	San Francisco, Cal.	0.615
Eugene, Ore.	0.439	Seattle, Wash.	0.340
Harrisburg, Pa.	0.495	Topeka, Kans.	0.612
Hartford, Conn.	0.532	Tulsa, Okla.	0.560
Helena, Mont.	0.521	Vicksburg, Miss.	0.447
Huron, S. D.	0.579	Wheeling, W. Va.	0.408
Indianapolis, Ind.	0.507	Wilmington, Del.	0.558

[a] To facilitate use of Fig. 5 the F values are based on a denominator for either latitude 35 deg or latitude 40 deg. Thus, whereas the actual latitude at Tampa, Fla., is 28 deg the F value is based on 35 deg. This increases rather than decreases the accuracy when used with Fig. 5.

in which,

∇_n = a solar gain term representing the reduction in excess normal transmission losses, such excess losses being $(U_s - U_w)(t_i - t_o')$, due to solar irradiation and expressed in Btu per hour. The term ∇_n must be numerically evaluated for each hour between sunrise and sunset of each day of the heating season.

r_o, r_a, r_g = thermal resistances of the outside air film, the air space and *each* sheet of glass, respectively.

E = the effectiveness factor of a solar window; equal to the fraction of the window which is unshaded at a particular hour of a particular day.

q_{a1} = the rate of solar energy absorption (at a given time and day) by the outside sheet of glass, but neglecting absorption of sky radiation.

$= I_o \cos i \, A_{(1 \text{ of } 2)}$

I_o = solar intensity on a plane normal to the earth's surface and at sea level; varies with time of day and day of month (refer to Fig. 1).

$\cos i$ = cosine of the angle of incidence of sun's rays on a window in a south wall (see Equation 30).

$A_{(1 \text{ of } 2)}$ = absorptivity of the outside sheet of two identical sheets of glass (refer to Equation 28).

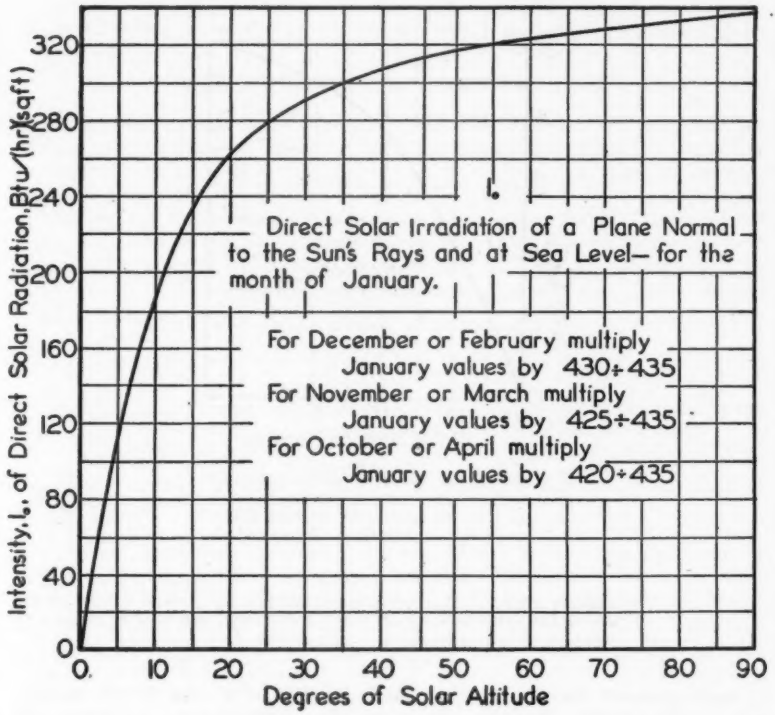

FIG. 1. DIRECT SOLAR IRRADIATION OF A PLANE NORMAL TO THE SUN'S RAYS AND AT SEA LEVEL FOR THE MONTH OF JANUARY

q_{a2} = the rate of solar energy absorption (at a given time and day) by the inside sheet of glass, but neglecting absorption of sky radiation.

$= I_o \cos i \, A_{(2 \text{ of } 2)}$

where

$A_{(2 \text{ of } 2)}$ = the absorptivity of the inside sheet of two identical sheets of glass (refer to Equation 29).

q_t = the rate of solar energy transmission (direct *plus* sky radiation at a given time and day) by the window.

$$= I_o \cos i \, T_{1,2} + I_o \frac{T_{s1,2}}{R}$$

where,

$T_{1,2}$ = transmissivity for direct solar radiation at a given time and day of a double glass window consisting of identical sheets of glass (refer to Equation 27).

$T_{s1,2}$ = transmissivity as just described but for sky radiation. This term can be taken as numerically equal to the value of $T_{1,2}$, which occurs at an angle of incidence of 60 deg.

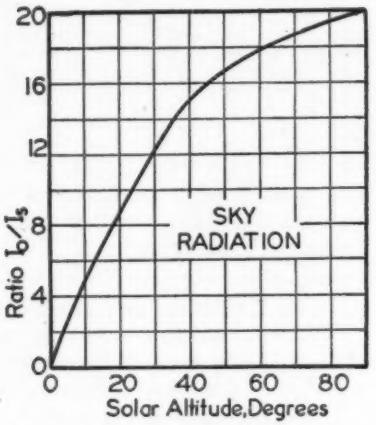

FIG. 2. RATIO OF INTENSITY OF DIRECT SOLAR RADIATION IN A PLANE NORMAL TO THE SUN TO INTENSITY OF SKY RADIATION ON A VERTICAL WINDOW

R = ratio of intensity of direct solar irradiation on a plane normal to the sun to intensity of sky radiation on a vertical window; R varies as a function of solar altitude as shown in Fig. 2.

Application of the foregoing rational equation to a solar house assumes two conditions: (1) That the rate of heat loss of the house as a whole will exceed the solar gain at all times during the heating season, or that for brief periods during which this condition is not true the thermal capacity of the floor will be sufficient to permit storage of excess solar energy for later re-use. (2) That all solar energy transmitted into the house eventually serves to reduce the heating plant output, for fixed inside air temperature, on a unit for unit basis. This condition implies effective insulation of directly irradiated surfaces.

Graphical solutions of this general equation are presented at the end of this paper for four varieties of double glass solar windows when used in either of two latitudes; use of these graphs makes it unnecessary for the designer to concern himself with variations in the solar intensity as a function

of time or season, or to calculate the angle of incidence. All results given here are for windows facing south.

Included in the integration is a term which takes account of the shading effect experienced during parts of the heating season from the roof overhang that is customarily used with solar construction. Development of additional graphical solutions for use with east and west walls and for localities where unusual atmospheric conditions exist is now underway and results will be reported in a later paper. Further, experimental study of solar heating is now being conducted on thermally and architecturally identical houses on the Purdue Housing Research Campus; results will be reported in the near future.

INSTANTANEOUS STEADY STATE CONDITIONS

The effectiveness for solar heating, of one square foot of glass depends at any given time on the net reduction (over that required for a corresponding square foot of wall area) in the net rate of heat flow from the heating plant to the surface of the window. Considering that all solar energy transmitted into the room eventually appears as an internal energy increase of the inside air and crediting the solar gain entirely to the glass, it is evident that at a steady state the actual transmission losses through the window will be equal to the sum of the solar transmission gain and the net energy supplied from the heating plant. This relationship can be written in equation form,

$$q_i = q_t + q_h \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (1)$$

where,

q_i = steady state transmission of heat from the interior of the room to the inside surface of the window. Btu/(hr) (sq ft of irradiated glass).

q_t = steady state transmission of solar energy into room through the window, Btu/(hr) (sq ft) of irradiated glass.

q_h = steady state makeup of energy from the heating plant required to maintain the inside air temperature predetermined value.

The most important qualification of Equation 1 arises from the assumed condition that solar transmission gains serve to reduce, unit for unit, actual transmission losses from the room. In order for this to be true all radiant energy entering through the window would have to be absorbed within the room and later released to the room air or reradiated (at long wave length) from a room surface. The first condition implies no direct loss through the window of reflected (from an interior room surface) solar radiation, whereas the second condition implies zero exposure (for perfect insulation) of the interim surface at which the solar radiation is absorbed; since by far the greater fraction of solar energy is received at the floor of the room it follows that the floor is assumed to be a non-transmitting surface. Note that for the problem now under consideration the thermal capacity of the floor is not an influencing factor since for a steady state condition there can be no passage of heat either to or from storage.

In order to solve Equation 1 for q_h the other two terms must first be evaluated. Heat transfer from the room to the inside window surface is given by,

$$q_i = h_i(t_i - t_{wi}) = q_t + q_h \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (2)$$

where t_i and t_{g1} are the temperatures of the room air and the inside surface of the window, respectively, whereas h_i is the combined coefficient for heat transfer by radiation and convection from the room to the window. Equation 2 is not yet subject to direct solution, however, as it contains two terms which need to be evaluated, q_h and t_{g1}; the latter term depends not only on the overall

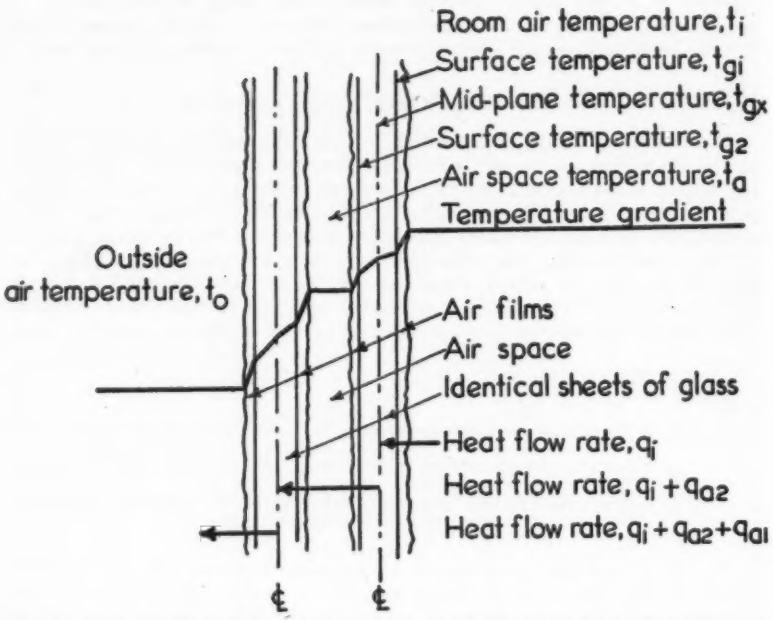

FIG. 3. HEAT BALANCE ON DOUBLE GLASS WINDOW FOR WINTER CONDITIONS WHEN WINDOW IS RECEIVING SOLAR RADIATION

coefficient of heat transfer for the window and on the outside air temperature, but also on the quantity of solar radiation absorbed by the glass.

Consider that the window consists of two identical thicknesses of glass separated by an air space (Fig. 3) the outside sheet absorbing q_{a1} and the inside sheet absorbing q_{a2} Btu/(hr) (sq ft). Then let r_g, r_a and r_o represent, respectively, the thermal resistances of one sheet of glass, of the separating air space and of the outside combined film coefficient. Let t_{gx} be the temperature at the midplane of the inner glass, t_a the airspace temperature and t_o the temperature of the outside air. The quantity of energy represented by q_{a2}, is absorbed almost uniformly throughout the thickness of the inner glass, but the analysis can be greatly simplified, without appreciable loss of accuracy, by treating this energy as though it were released from a heat source located

in the midplane of the inner glass. The temperature gradient from inside surface to the midplane would, therefore, be a straight line (neglecting the variation of thermal conductivity with temperature) based on transmission at a rate of q_1 while the gradient from the midplane to the outer surface of the inner glass would be a steeper straight line based on the transmission at a rate of $q_1 + q_{a2}$ Btu/(hr) (sq ft). An equation can be written for heat transfer through the inner half of the inner glass,

$$q_1 = \frac{t_{g1} - t_{gx}}{(r_g/2)} = \frac{2(t_{g1} - t_{gx})}{r_g} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

and a similar equation for transmission through the outer half of the inner glass,

$$q_1 + q_{a2} = \frac{t_{gx} - t_{g2}}{(r_g/2)} = \frac{2(t_{gx} - t_{g2})}{r_g} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3a)$$

where t_{g2} is the surface temperature of the outside of the inner glass. Adding Equations 3 and 3*a*,

$$2q_1 + q_{a2} = \frac{2(t_{g1} - t_{g2})}{r_g} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

An equation can also be written (similar to Equation 2) for the heat transfer from the outside surface of the inner glass,

$$q_1 + q_{a2} = \frac{t_{go} - t_a}{r_a/2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

Substituting the equivalent inside film resistance, r_i, in Equation 2 places it in the same form as Equation 5,

$$q_1 = (t_i - t_{gi})/r_i \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

Equations 4, 5 and 6 can now be solved simultaneously to eliminate the two unknown surface temperatures of the inner sheet of glass. From Equation 4,

$$t_{gi} = \frac{r_g}{2}\left(2q_1 + q_{a2} + \frac{2t_{go}}{r_g}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4a)$$

and from Equation 6,

$$t_{gi} = (h_i t_i - q_1)/h_i \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6a)$$

Equating 4*a* and 6*a* and reducing,

$$2h_i t_i - 2q_1 = 2r_g h_i q_1 + h_i r_g q_{a2} + 2h_i t_{go}$$

and solving for t_{go},

$$t_{go} = t_i - q_1(r_i + r_g) - \frac{r_g}{2}q_{a2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (7)$$

But from Equation 5,

$$t_{go} = \frac{r_a}{2}(q_1 + q_{a2}) + t_a \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5a)$$

Eliminating t_{go} between equations 7 and 5*a* and reducing,

$$q_i = \left[\frac{(t_i - t_a) - \left(\frac{r_g}{2} + \frac{r_a}{2}\right)}{\frac{r_a}{2} + r_g + r_i} \right] q_{a2} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (8)$$

But,

$$\frac{1}{\frac{r_a}{2} + r_g + r_i} = U_{g2} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (9)$$

where U_{g2} is the overall coefficient of heat transfer from room air to air space. Then substituting in Equation 8,

$$q_i = U_{g2}(t_i - t_a) - U_{g2}\left(\frac{r_g}{2} + \frac{r_a}{2}\right) q_{a2} \quad \ldots \ldots \ldots \ldots \ldots \quad (10)$$

Note that the term $\frac{r_g}{2} + \frac{r_a}{2}$ in the above Equation 10, is equal to the thermal resistance from the midplane of the inner glass to the middle of the air space.

Equation 10 is not in a form satisfactory for use because it includes the unknown air space temperature, t_a. This term can be eliminated by writing an equation analogous to Equation 10 for the outer of the two sheets of glass, then solving the two equations simultaneously. Referring to Fig. 3, an equation similar to Equation 10 is,

$$q_i + q_{a2} = U_{g1}(t_a - t_o) - U_{g1}\left(\frac{r_g}{2} + r_o\right) q_{a1} \quad \ldots \ldots \ldots \ldots \quad (11)$$

Solving Equation 10 for t_a,

$$t_a = \frac{\left[U_{g2} t_i - U_{g2}\left(\frac{r_g}{2} + \frac{r_a}{2}\right) q_{a2} - q_i \right]}{U_{g2}} \quad \ldots \ldots \ldots \ldots \quad (10a)$$

and from Equation 11,

$$t_a = \frac{\left[U_{g1} t_o + U_{g1}\left(\frac{r_g}{2} + r_o\right) q_{a1} + q_i + q_{a2} \right]}{U_{g1}} \quad \ldots \ldots \ldots \quad (11a)$$

Eliminating t_a between Equations 10*a* and 11*a* and reducing,

$$t_i - \left(\frac{r_g}{2} + \frac{r_a}{2}\right) q_{a2} - \frac{q_i}{U_{g2}} = \frac{(q_i + q_{a2})}{U_{g1}} + \left(\frac{r_g}{2} + r_o\right) q_{a1} + t_o \quad \ldots \ldots \quad (12)$$

or,

$$q_i = \left\{ t_i - \left(\frac{r_g}{2} + \frac{r_a}{2}\right) q_{a2} - \frac{q_{a2}}{U_{g1}} - \left(\frac{r_g}{2} + r_o\right) q_{a1} - t_o \right\} \frac{1}{\frac{1}{U_{g1}} + \frac{1}{U_{g2}}} \quad \ldots \quad (13)$$

but,

$$U_c = \frac{1}{\frac{1}{U_{g1}} + \frac{1}{U_{g2}}} \quad \ldots \ldots \ldots

instantaneous rate of saving in energy requirements (for fixed solar conditions) would be the same for zero weather as for warm weather.

SEASONAL CONDITIONS

Equation 19 was developed for determining the instantaneous rate of saving corresponding to any given set of steady state conditions. Practical application of this equation would require that it be adapted to the determination of the actual quantitative saving expressed in Btu per heating season. Thus,

$$\Delta Q_{max} = \int_{season} E(\Delta q)d\theta = \sum_{season} E\nabla(\Delta\theta) - (U_c - U_w)\sum_{season}(t_i - t_o)\Delta\theta \quad . . (20)$$

where $\Delta\theta$ = a small increment of time

E = an *effectiveness factor* which accounts for partial shading (due to roof overhang) of the window.

The summation of Equation 20 is to be carried out over the entire heating season and the resultant ΔQ_{max} will be the saving available in a locality where the sun is never obscured.

If $\Delta\theta$ is taken as 1 hour and the heating season is considered to extend from October 1 to May 1 the number of $\Delta\theta$ intervals would be 5088 and Equation 20 would become,

$$\Delta Q_{max} = \sum_{n=1}^{n=5088} \nabla_n - (U_c - U_w) \sum_{n=1}^{n=5088} (t_i - t_{on}) \quad . . . . . (21)$$

where $\nabla_n = E\nabla'$ . . . . . (21*a*)

A term t_o' can now be defined by the equation,

$$t_i - t_o' = \sum_{n=1}^{n=5088} \frac{(t_i - t_{on})}{5088} \quad . . . . . (22)$$

giving, $t_o' = \sum_{n=1}^{n=5088} \frac{t_{on}}{5088}$ . . . . . (22*a*)

Thus t_o' is the *average* outside air temperature during the heating season and its approximate value for any locality can be obtained from Table 1 (pp. 258-9) of the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1946. Equation 21 therefore simplifies to the form,

$$\Delta Q_{max} = \sum_{n=1}^{n=5088} \nabla_n - 5088(U_c - U_w)(t_i - t_o') \quad . . . . . (23)$$

In its present form Equation 23 expresses the seasonal saving, in Btu, that would be attributable to 1 sq ft of glass if the sun were to shine for 100 per cent of the possible sunshine hours of the heating season. During the seven month heating period considered in this paper the maximum possible hours of sunshine (determined from data on the time of sunrise and sunset) are 2278 hours for latitude 40 deg and 2360 for latitude 35 deg. For many

localities the weather bureau can provide information on the average number of actual hours of sunshine to be expected during a particular month and from this one can readily calculate the fraction of maximum possible time, for any locality, that the sun will be expected to shine. Table 1 gives fractions for a large number of American cities; values in the table were computed by dividing the average number of sunshine hours during a seven month heating season by the maximum possible hours as stated above; data on actual hours are taken from a recent article in *Heating and Ventilating*[1] which was developed from a weather bureau report.[2] The sunshine fraction is defined as the *usage ratio* and identified by the symbol F. With this substitution Equation 23 can now be written in a form which will give the *actual* seasonal saving in any locality,

$$\Delta Q = F \sum_{n=1}^{n=5088} \nabla_n - 5088(U_e - U_w)(t_i - t_o') \quad \ldots\ldots\ldots \quad (23a)$$

If complete data were available on the distribution of actual sunshine hours through the day (for a given locality) the above Equation *23a* could be given added accuracy by placing F under the summation sign and letting it represent the statistical fraction for a given hour rather than for the entire heating season. As the equation is now written the fraction of sunshine is necessarily treated as though it were the same for each hour of the day. Although such a uniform distribution does not actually occur, it is probable that for most localities the error introduced from this cause is not great.

Before proceeding with the evaluation of Equation *23a*, a not inconsiderable problem remains; that is, to determine the seasonal solar effect as expressed in the summation $\Sigma\nabla_n$.

THE SOLAR INCREMENT: INSTANTANEOUS VALUE

The solar term, defined in Equation *21a*, relates three characteristics of the system; these are: (1) the solar transmission and absorption rates q_t, q_{a1}, q_{a2}; (2) the *effectiveness ratio*, E, which is defined as the fraction of the window which, at a given time, is not in the shade; (3) the thermal characteristics of the particular window as identified by the terms K_1 and K_2 *where*,

$$K_1 = U_e(r_o + 1.5r_g + r_a) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (24)$$

$$K_2 = U_e\left(\frac{r_a}{2} + \frac{r_g}{2}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (25)$$

giving (by substitution into Equations 18*a* and 21*a*),

$$\nabla = E(K_1 q_{a1} + K_2 q_{a2} + q_t) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (26)$$

For a given type of window K_1 and K_2 have fixed values so ∇ is then a function only of E and of the three solar energy rates. Each of these rates depends on the physical characteristics of the glass, the angle of incidence

[1] Heat Gain From Winter Sunshine in the U. S., *Heating and Ventilating*, March 1946, pp. 63-66.
[2] *Atlas of American Agriculture*, Section B, Joseph B. Kincer, U. S. Weather Bureau.

and the solar irradiation. Each of these factors will be considered in some detail:

Glass Characteristics. When a sheet of glass is irradiated some of the energy is reflected, some absorbed and some transmitted. The characteristics of the glass which determine the distribution between these three forms of energy disposal are the thickness and the absorption coefficient, their product, KL, being the determining factor. For the purposes of this paper the only two kinds of glass considered were double strength A quality—with a KL value of 0.025—and clear ¼ in. plate glass with a KL of approximately 0.050. Using equations available in the literature,[3] the absorptivity, reflectivity and transmissivity were each determined as a function of angle of incidence. Likewise, the transmissivity was determined for windows made up of two identical sheets of glass, each sheet having a KL value of either 0.025 or 0.505 and the absorptivity determined for both the inner sheet and the outer sheet of these windows. Equations used for the window characteristics are modified forms of the equations derived in Parmélee's paper,[3]

$$T_{1,2} = T_1^2/(1 - R_1^2) \quad \ldots \quad (27)$$

$$A_{1 \text{ of } 2} = (1 - R_1^2 + R_1 T_1)\left(\frac{A_1}{(1 - R_1^2)}\right) \quad \ldots \quad (28)$$

$$A_{2 \text{ of } 2} = \left(\frac{(T_1 A_1)}{(1 - R_1^2)}\right) \quad \ldots \quad (29)$$

where T_1, A_1, and R_1 represent transmissivity, absorptivity and reflectivity, respectively; the subscript 1,2 indicates that the term is for the combination of two sheets of glass, whereas subscripts 1 of 2 and 2 of 2 indentify the property as valid for the first of two sheets and the second of two sheets respectively; each of these properties varies as a function of the angle of incidence.

The Angle of Incidence, i, depends on the position of the sun and must, therefore, vary continuously throughout the heating season. At any given time this angle can be calculated from the solar altitude, the sun's azimuth and the orientation of the window. In this paper only windows located in walls facing south are considered and for this case the angle of incidence can be calculated from the equation,

$$\cos i = \cos H \cos (180 - A) \quad \ldots \quad (30)$$

or,
$$i = \cos^{-1} [\cos H \cos (180 - A)] \quad \ldots \quad (30a)$$

where i = angle of incidence of direct solar irradiation (measured from a normal to the window).

H = solar angle (from elevation of the sun above the horizon) determined from *Hydrographic Office Bulletin* 214.

A = sun's azimuth; measured in degrees east or west of north; determined from *Hydrographic Office Bulletins* 214 and 71.

The Solar Intensity. The total solar intensity experienced on any vertical surface consists in part of an intensity, I_v, representing the component of direct

[3] A.S.H.V.E. Research Report No.1281—The Transmission of Solar Radiation Through Flat Glass Under Summer Conditions, by George V. Parmelee. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 317.)

solar radiation impinging on the surface and in part of an intensity, I_s, representing sky radiation and consisting largely of solar radiation which was scattered by dust and moisture during passage through the earth's atmosphere and is received diffusely at the surface rather than, as for direct radiation, as a solar beam. Because it is received from all directions, the transmissivity and absorptivity values applicable to sky radiation are independent of time and of latitude and have a fixed value for each particular type of window. Based on a graphical integration for diffuse irradiation, Parmelee [4] states that the integated values of A_s and T_s can be considered equivalent to the specular values based on an incidence angle of 60 deg. Then for double glass windows, the absorptivity with respect to sky radiation is,

$$A_{s(1\text{ of }2)} = A_{1\text{ of }2} \text{ (for 60 deg)} \quad \ldots \quad (31)$$

and

$$A_{s(2\text{ of }2)} = A_{2\text{ of }2} \text{ (for 60 deg)} \quad \ldots \quad (31a)$$

while the transmissivity of the combined two sheets of glass is,

$$T_{s1,2} = T_{1,2} \text{ (for 60 deg)} \quad \ldots \quad (31b)$$

If, now, the q terms of Equation 26 are separated into direct solar and sky radiation fractions and are then replaced by their respective products of intensity and of absorptivity or transmissivity, the result is,

$$\nabla = E[K_1(I_v A_{1\text{ of }2} + I_s A_{s(1\text{ of }2)}) + K_2(I_v A_{(2\text{ of }2)} + I_s A_{s(2\text{ of }2)}) + (I_v T_{1,2} + I_s T_{s1,2})] \quad \ldots \quad (32)$$

where,

I_v = intensity of direct solar irradiation received on each square foot of irradiated window area,

I_s = intensity of sky radiation received on each square foot of irradiated window area.

Detailed attention must now be given to procedures for obtaining the instantaneous value of I_v and I_s.

Direct Solar Intensity. Basic to any discussion of solar effects is a knowledge of the irradiation intensity, expressed in Btu/(hr) (sq ft) on a plane normal to the sun's rays and located on the earth's surface. This intensity depends on the length of the path through the earth's atmosphere (determined by solar altitude), the character of the atmosphere (determined by dust concentration,[5] ozone concentration and the quantity of water vapor [6]) and on variations in the intensity of solar radiation outside the earth's atmosphere. Extra atmospheric variation is due principally to seasonal variation in the distance of the earth from the sun. In this paper the variation has been taken as five units per month from a base value of the solar constant of 435 Btu/(hr) (sq ft) in January; the correction is subtracted both for the months that follow and for those that precede January.

Variations in length of path through the atmosphere can be readily expressed in terms of the solar altitude.

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[4] Loc. Cit. Note 3.

[5] Review of U. S. Weather Bureau Solar Radiation Investigations, by Irving F. Hand. (*Monthly Weather Review*, Vol. 65, 1937, Table 12, pp. 441.)
Proposed Standard Solar Radiation Curves for Engineering Use, by Parry Moon. (*Journal of the Franklin Institute*, Vol. 230, No. 5, November 1940, p. 605.)
Turbidities of American Air Masses and Conclusions Regarding Seasonal Variation in Atmospheric Dust, by C. H. Wexler. (*Monthly Weather Review*, Vol. 62, 1934, pp. 397-402.)

[6] *Monthly Weather Review, Climatological Data Section*, Vol. 73, No. 1, Jan. 1945.

Variation of intensity at fixed solar altitude depends on the absorption characteristic of ozone in the earth's atmosphere, the scattering effect due to dust and to water vapor, the absorption effect of water vapor and, to a lesser extent, to the absorption effect of carbon dioxide. Analytical means of determining the intensity corresponding to a given dust and vapor concentration are available in the literature[7] and where extremes of either humidity or dustiness (as most commonly evidenced by smokiness in the vicinity of indus-

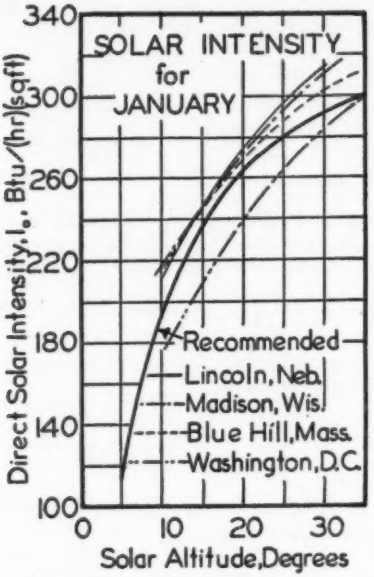

FIG. 4. REPRESENTATIVE EXPERIMENTAL VALUES OF DIRECT SOLAR INTENSITY VS SOLAR ALTITUDE

trial areas) are experienced a special solar analysis, for known local conditions, should be undertaken; in a subsequent paper the authors hope to develop graphical solutions of the solar problem which will include both water vapor and dust as variables and will thereby permit the designer to select conditions corresponding to the actual atmosphere with which he is concerned. For the present, however, a procedure has been followed which is less exact, but is adequate for engineering purposes in most parts of the continental United States. Exceptions not subject to accurate treatment would occur in localities at high elevation or with very low or very high average winter dew points.

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[7] Proposed Standard Solar Radiation Curves for Engineering Use, by Parry Moon. *Journal of the Franklin Institute*, Vol. 230, No. 5, Nov. 1940, p. 605.

The recommended curve for direct solar intensity as a function of solar altitude is given in Fig. 1; similar curves for other months can be obtained by multiplying the intensity for January by the ratio of the solar constant for the month in question to the solar constant for January. The recommended curve was developed from theory [8] for a dust concentration of 300 particles per cubic centimeter (particles assumed uniformly 1 micron in size) and a vapor content corresponding to 33 F dew point temperature. The selected dust concentration is the same as that used by Moon [8] in developing his recommended intensity curve, but the dew point is substantially lower than the value used by Moon. This difference arises from the fact that Moon's curve is intended for summer use in calculating cooling load whereas the recommended curve of Fig. 1 was specifically selected for use over the seven month heating season. Aside entirely from the theoretical considerations which entered into its development, the curve justifies itself by representing an effective, single line correlation of experimental intensity determinations. Representative experimental curves are shown in Fig. 4 which is for January and hence has 30 deg as the maximum solar altitude.

The solar intensity I_o, given in Fig. 1, is for direct irradiation on a plane normal to the sun's rays. To obtain the irradiation, I_v, the normal intensity must be reduced to compensate for the greater vertical area corresponding to unit normal projected area. Thus,

$$I_v = I_o \cos i \quad \ldots \ldots \ldots

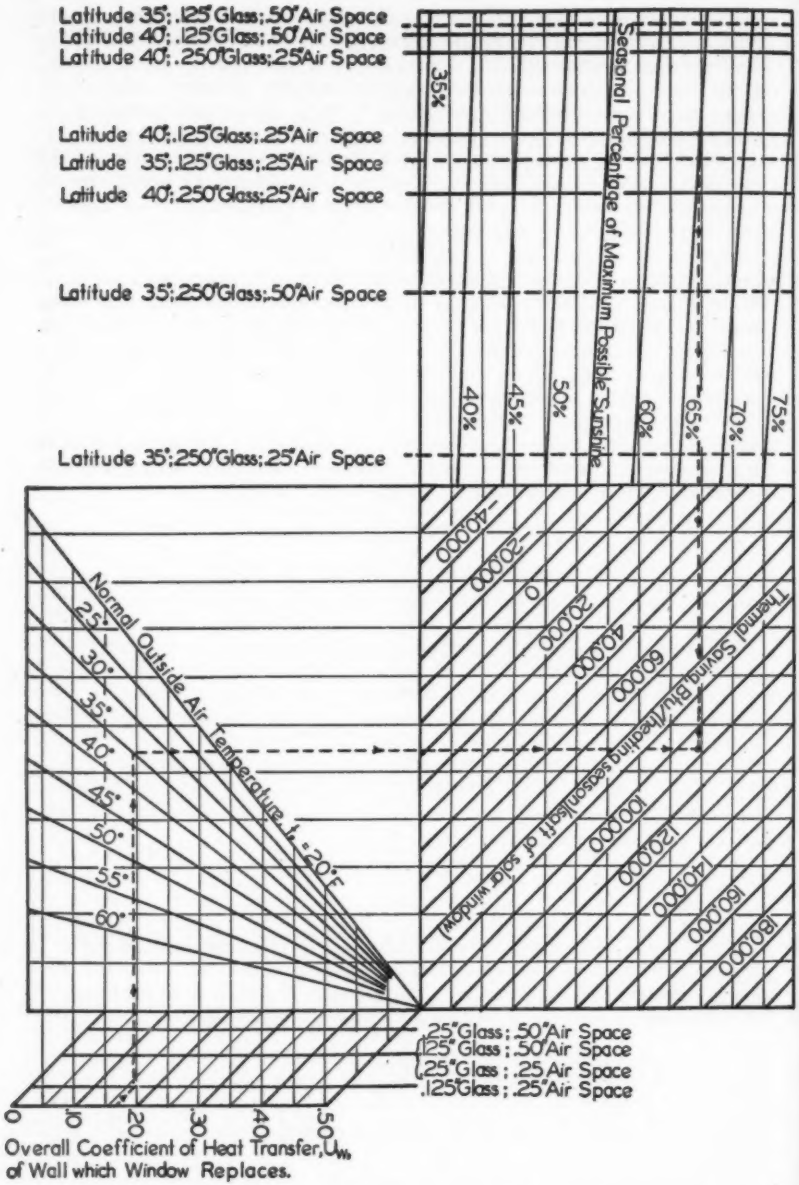

FIG. 5. SEASONAL SAVING (OR LOSS) ATTRIBUTABLE TO 1 SQ FT OF DOUBLE GLASS WINDOW REPLACING AN EQUAL AREA OF SOUTH WALL WITH SOLAR OVERHANG OF ROOF

(Note that the opaque wall area is not credited with any effective seasonal solar energy utilization)

Equation 35 is difficult to work with. Calculation of numerical values of ∇ over a wide range of i shows that, *for the heating season,* neglect of the absorption fraction of sky radiation (but retaining the T_s term) will introduce an error in ∇ of approximately ½ of 1 per cent. This value is too small to be of practical significance hence Equation 35 can, with accuracy, be simplified to the form,

$$\nabla = E(I_o \cos i)\left[K_1 A_{1 \text{ of } 2} + K_2 A_{2 \text{ of } 2} + T_{1,2} + \frac{T_{s1,2}}{R \cos i}\right] \quad \ldots\ldots (36)$$

$$= E(I_o \cos i)\phi = EI_v\phi \quad \ldots\ldots\ldots\ldots\ldots\ldots (37)$$

where,

$$\phi = K_1 A_{1 \text{ of } 2} + K_2 A_{2 \text{ of } 2} + T_{1,2} + \frac{T_{s1,2}}{R \cos i} \quad \ldots\ldots\ldots\ldots (38)$$

The Effectiveness Ratio, E. In designing a building to permit maximum utilization of solar energy during the heating season a parallel requirement is to provide maximum exclusion of solar energy during the cooling season. Unfortunately these two design objectives are, in large measure, mutually irreconcilable. If sufficient roof overhang is provided to permit shading of the entire window at solar noon on June 21 the net result will inevitably be to shade a substantial fraction of the window at solar noon in February and in October, two months during which heating requirements may indicate the need of obtaining maximum solar irradiation. With any effective use of roof overhang there will necessarily be a reduction in available solar heating effect since a fraction of the window will often be in shadow.

In order to take account of this shading effect the square foot of glass used as a basis for the analytical treatment of this paper is considered to be a vertical strip of length depending on the amount of roof overhang, so located as to be fully in the shade at solar noon on June 21, fully in the sun on December 21 and partially shaded at solar noon on all other days of the year. Both the dimensions of the strip and the height above the floor of the window sill will be fixed for any given amount of overhang, but for all windows which meet this prescribed condition the fraction of glass area in the shade at a particular day, time and latitude will necessarily be the same irrespective of window shape or area. This fraction, the E term, can be calculated—for any latitude, day and time—from a knowledge of the solar altitudes for that latitude at the day and time in question and at solar noon on December 21 and June 21.

THE SEASONAL ADVANTAGE (OR LOSS) DUE TO SOLAR HEATING

All required terms can now be evaluated and Equation 23*a* can, therefore, be solved by calculating the 5088 values of ∇_n which are needed to permit a seasonal summation. Formidable as this task may seem, it is not unduly difficult since more than half of the ∇_n terms will necessarily be equal to zero. The number of terms requiring evaluation is, therefore, reduced to something under 2500, the exact number varying with latitude and being numerically equal to the maximum possible hours of sunshine during the seven month heating season. Corresponding to latitudes 35 deg and 40 deg the maximum number of ∇_n terms are 2360 and 2278 respectively.

A further reduction can be attained by selecting one date as representative of average conditions during an entire month and by noting further that the average solar days of October and February are identical as are the average solar days of November and January. Thus a solar analysis of five days of the heating season, each day having approximately 10 possible hours of sunshine, will give data from which the 2360 ∇_n terms (for latitude 40 deg) can be evaluated. Even so, the task of calculating approximately 50 values of ∇_n for each latitude and for each type of window at each latitude is no small one. The results presented in this paper are for two latitudes and four types of window, hence required computation of 400 basic ∇_n terms, including approximately 10,000 unit calculations.

Fig. 5 is a graphical solution of Equation 23*a* based on values of $\sum_{n=1}^{n=5088} \nabla_n$

which have been determined by tabular integration using the procedure developed in earlier sections of this paper. The solution given in Fig. 5 is as accurate as the assumptions which underlie the method.

Example: A window consisting of two identical sheets of ⅛ in. clear plate glass separated by a ¼ in. air space is to be used in a south wall which has an overall coefficient of heat transfer of 0.165 Btu/(hr) (sq ft) (F). The normal outside temperature for the locality is 35 F and the sun shines for 65 per cent of the maximum possible hours between October 1 and May 1. Latitude is 40 degrees.

Solution: Enter bottom of the upper left quadrant of Fig. 5 at the heavy vertical line which is identified as applicable to ⅛ in. glass with ¼ in. air space at 40 deg latitude. Rise along this line to intersection with line for 65 per cent sunshine then move horizontally right (see dashed example line) to enter upper right quadrant. Now re-enter the figure at value of $U_w = 0.165$ on scale at right of lower right quadrant. Follow the directrix line from point of entry to intersection with vertical for ⅛ in. glass, ¼ in. air space, 40 deg latitude; then move horizontally left to intersect curve for $t_o = 35$ deg and from this point move vertically upward to intersect the horizontal line already established in the upper right quadrant. The point of intersection of these two lines gives the answer as 107,000 Btu saved per seven month heating season per square foot of window.

SUMMARY

The gain (or loss) resulting from use of unit area of glass in place of unit area of wall is developed in terms of a rational general equation which gives the result in terms of the Btu saving per square foot of glass per heating season. The period adopted as a standard heating season is from October 1 to May 1. This choice permits determination from the HEATING, VENTILATING, AIR CONDITIONING GUIDE of the normal outside temperature for this period in most parts of the United States. Table 1, based on Weather Bureau data, is included to give normal percentage of maximum obtainable sunshine that is experienced in 48 American cities during the heating period indicated.

Integrations of solar effects over the entire heating season are carried out for two latitudes and four combinations of double glass with separating air

space in solar windows; all cases considered are for windows located in walls facing south. Results of the analysis are shown graphically in Fig. 5; from this figure one can determine the seasonal gain (or loss) without calculation. Additional constants for use with east and west walls and in localities where atmospheric conditions are very unusual are being developed and will be presented, together with experimental results, in a later paper.

DISCUSSION

G. L. TUVE, Cleveland, Ohio: The Society has received many requests for information on radiant heating and solar heating. The various steps required in a research program relating to these subjects is outlined in this paper.

G. LORNE WIGGS, Montreal, Canada: Were the two panes of glass forming the air space hermetically sealed and can the graphical solution be applied to double glass whether air is contained between the panes or not?

C. F. KAYAN, New York, N. Y.: I would be interested in the authors' comments on the day and night temperatures obtained in the solar and non-solar houses.

C. O. MACKEY, Ithaca, N. Y.: The author is to be complimented on a study which is extremely important. How did the authors obtain the data for the intensity of solar and sky radiation on a vertical surface? Was it obtained from Weather Bureau data, or was it simply calculated as if radiation were all beamed and then corrected as beamed radiation through trigonometrical means for vertical surfaces?

Does the author propose to make similar calculations for other orientations than the principal one, since obviously there might be optimum orientation such as 15 deg west of south. Also, will the charts in this paper apply to a single glass as well as a double glass construction?

W. E. HEIBEL, New York, N. Y.: Does the type of glass used have any effect in filtering the solar rays and are some types of glass better than others?

W. A. DANIELSON, Memphis, Tenn.: In living in a solar type house I have found it desirable to draw curtains at night to prevent heat loss.

L. P. HYNES, Philadelphia, Pa.: Have any tests been made using glass in an inclined position such as found in greenhouses?

PROFESSOR HUTCHINSON: The graphical solution applies regardless of the gas or air contained between panes of glass, provided large quantities of CO_2 or water vapor are not present. During the night the temperature in the solar house was lower than in the non-solar type. During the day there was a rise in temperature in the solar house even when there was no sunshine. It is also of interest to know that the temperature of the non-solar house exceeded the outside temperatures, thereby showing that all houses receive some solar heat.

Referring to Professor Mackey's questions, the solar intensity curve includes all forms of radiation except sky radiation. Allowance was made for sky radiation insofar as it represented energy transmitted through windows.

Orientation is extremely important and will be considered in further experiments.

In answer to Mr. Heibel's inquiry in summer it is desirable that glass transmit a minimum of heat and a maximum of light, but in winter for solar heating purposes any clear glass would be adequate. Solar heat loss into the building was

independent of outside temperature, but the heat loss from the room through the windows to the outside increased with a decrease of outside temperature.

No tests of glass in an inclined position have been planned under the present appropriation of funds.

Referring to General Danielson's remark, the tests showed results for all the solar energy that could pass into the house, but obviously results would be different in a house where drapes were closed at night to keep heat in and where they might also be closed during part of the day.

1302

No. 1302

HOLD-FIRE CONTROLS FOR BITUMINOUS COAL STOKERS †

By C. C. WRIGHT,* T. S. SPICER,** AND H. A. BAUMANN ††

ONE of the major considerations associated with the satisfactory operation of the domestic stoker during mild weather is the ability of the stoker to maintain fire without causing any overheating of the home. This self-maintenance of ignition in solid fuel burning equipment is essential to satisfactory performance because no automatic ignition system, such as commonly used with oil and gas burning equipment, has yet been developed for coal stokers. During the summer, when the stoker may be used only for service water production, hold-fire control is a major factor in satisfactory performance because operation of the stoker on heat demand should occur only after major draw-offs of service water and these rarely occur more than once or twice a day. At all other times, the hold-fire rate should be equal to or less than the radiation losses of the boiler, piping, and hot water storage tank, otherwise some overheating of the home or loss of efficiency occurs.

The heat losses from the boiler, piping, and storage system will depend largely upon the nature of the system and the type of insulation provided. In home tests [1] completed some time ago on the production of service water in a stoker fired boiler, it was estimated that these losses for the particular system (insulated boiler, but uninsulated piping and storage tank) were equivalent to a coal consumption of approximately 0.7 lb per hour. More recent test data on the laboratory installation in which hold-fire tests were conducted indicate that this figure may be somewhat high for the average system and will more nearly approach 0.5 lb per hour. On the basis of these estimates, therefore, it appears reasonably certain that with a properly insulated system

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† A contribution from the Division of Fuel Technology of The Pennsylvania State College, State College, Pa. This investigation is part of a major program on Comfort Heating with Bituminous Coal and was sponsored jointly by the Commonwealth of Pennsylvania through its Department of Mines, the *Western Pennsylvania Coal Operators Association* and the *Central Pennsylvania Coal Producers Association.*

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†† Research Assistant, Department of Fuel Technology, The Pennsylvania State College.

[1] Exponent numerals refer to bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Montreal, Que., Canada, June 1946.

a hold-fire rate of 0.4 to 0.6 lb of coal per hour will maintain service water temperature except during draw-off periods.

BASIC PRINCIPLES OF HOLD-FIRE OPERATION

During winter operation, there is little or no demand for hold-fire operation except possibly during the night, when day-night thermostats are used because the stoker operates primarily according to the thermostatic demand. During the remainder of the year, however, the hold-fire control is frequently the operating control. An examination of the conditions of hold-fire operation during different types of demand indicate that there are two basically different conditions that may prevail.

During mild weather, such as is common during early fall and late spring, the hold-fire control functions as the operating control for a few hours each day, after which the heat demand (thermostat) control again becomes the operating control. Depending upon the characteristics of the stoker and the coking properties of the coal, the fairly heavy firing periods occurring during the early morning tend to build up a fuel bed of from 5 to 20 lb of coal and coke which acts as a reserve during the subsequent hold-fire period. Although in terms of quantity this reserve of fuel is usually adequate to maintain the fire for several hours, some form of hold-fire control is necessary to insure periodic operation of the stoker and maintain ignition of this fuel.

During the summer, when the stoker boiler combination is used for service water production, the hold-fire control functions as the primary operating control for long periods of time with only short and infrequent heat (aquastat) demand periods interspersed. Under these conditions little or no reserve of fuel is present in the firebox and the fire is maintained on the coal fed during the hold-fire operating periods.

HOLD-FIRE EQUIPMENT AVAILABLE

There are in use two basically different classes of hold-fire equipment. The more commonly used is the time control which functions on the principle of periodic operation at predetermined time intervals. The second class depends for its operation on changes in temperature affecting a temperature sensitive element which in turn actuates the operation of the stoker.

Time Controls

There are numerous time lapse hold-fire devices available, including the clock, electrical resistance and hydraulic types, having various degrees of refinement and accuracy; the general principles of their design are fairly well known and require no elaboration. With this class of control it is possible to set the time lapse mechanism so that it will operate the stoker for a predetermined number of minutes per cycle. In the early controls, this cycle was usually one hour, but with all but the most freely burning bituminous coals it was found that a one hour cycle was usually inadequate and, therefore, most of the controls now used for bituminous coal stokers can be set to operate on at least a half hour cycle while several operate on one quarter hour cycles. In the better type of time lapse controls, the hold-fire control is automatically

reset to zero at the end of an operating period so that the hold-fire cycle starts from this point.

It is frequently claimed that with free burning coals this type of control will maintain fire on an operating cycle of 1½ to 2 min per hour or in terms of coal fed, 0.5 to 0.6 lb per hour. No reliable test data appear to have been published which substantiate this claim and the experience of the authors based for the most part on the use of coking coals would indicate about 1 lb per hour as the minimum and 1.5 to 2.0 lb per hour as the average. Field surveys in the area where strongly coking coals are normally used indicate that 3 to 4 lb per hour is not an uncommon requirement to insure maintenance of ignition and that 15 or 30 min cycles are almost essential.

Temperature Controls

The other major class of hold-fire control is actuated not on a time lapse basis, but rather by changes in temperature of either the firebox or the stack gases. In general, the stack temperature type has not proved satisfactory for coal fired units because the control response bears little relation to the need of the fire for attention and the only temperature actuated units now used extensively are those whose operation depends upon fire-box temperature.

In this type of control, a heat responsive element is inserted into the firebox. When the temperature of the firebox falls below some preset minimum, the element actuates the control, causing the stoker to operate until the firebox temperature rises to some preset maximum, at which point the control stops the stoker operation.

Since this class of control depends only on firebox temperature, it should not only be much more reliable for maintaining the fire and preventing overheating of the home, but should also be much more efficient, because the only calls for operation will occur when the fire actually needs attention. In home tests,[1, 2] it has been found possible, when using this type of control, to obtain satisfactory performance at a hold-fire rate of 1.3 to 1.5 lb per hour even with some of the strongly coking coals, while in the laboratory on prolonged hold-fire operation with no interspersed heat demands, satisfactory operation was maintained for two weeks at a rate of 0.7 lb per hour on a similar coal.

When used in the present type of furnace or boiler fired with the clinkering type stoker, this type of control possesses the disadvantage of being located in the firebox, usually inserted through the fire door, where it may be damaged by careless use of the clinker tongs. When used with automatic ash removal stokers, however, no such disadvantage exists. In the design of future furnaces and boilers intended for stoker firing, it is to be hoped that some provision will be made for insertion of the temperature type control in a more convenient and less exposed location. Unfortunately, the only control of this type available prior to the war was so constructed mechanically that the heat element was subjected to load when the fire was hottest and despite use of alloy material in construction, the control did lose adjustment and sensitivity. Experience with a number of these units in home and laboratory tests indicated their useful life, even when readjusted each year, was only about 3 to 5 years and, where abnormally high firebox temperatures were encountered, as short as a full heating season.

In consideration of test results discussed later and from experience with a variety of hold-fire controls, the authors outlined in 1942 the requirements for an improved control which, it was believed, would overcome a number of the deficiencies of existing units. These requirements, later submitted to a number of control manufacturers, were as follows:

1. A temperature actuated control is highly desirable.

2. The unit must be so constructed that the heat responsive element does not lose adjustment or sensitivity because of load at high temperatures (up to 1400 F) or for other reasons.

3. The unit should be so designed that when changing from a firing cycle to hold-fire operation, the control will have some mechanism for increasing the hold-fire rate during the transition period so that ignition will not be lost due to more rapid cooling of the fire than of the furnace refractories or castings.

4. The unit should have a built-in outfire control to prevent feeding of coal in case the fire does for any reason go out. Failure of electric power supply during the summer months may occur in certain localities and, if prolonged, may cause the fire to go out.

5. The unit should be adjustable for both cut-on (minimum) temperature and differential so that it will be applicable for all classes and sizes of furnaces and boilers.

6. The frequency of operation of the control on normal hold-fire operation should not be oftener than about four times per hour.

7. Mercury type toggle switches should be eliminated, if possible, because of their tendency to cause radio interference under certain conditions, which would prove objectionable in the home.

8. The unit must be competitive in price range with the better quality controls now marketed.

A control meeting the foregoing specifications has been designed and is under laboratory and field tests. This unit, the basic elements of which are shown diagrammatically in Fig. 1, has been subjected to extensive laboratory tests by the authors during the past year and has so far fulfilled all the requirements outlined, except that insufficient data are available as yet to establish the life of the unit.

The general principles of operation of the control shown diagrammatically in Fig. 1 are as follows:

During normal hold-fire operation the moving arm C, to which is attached arm C′ carrying the clutch cam H, is in the second or rear notch on cam H. As the firebox temperature falls, the heat responsive element A causes the moving arm C together with arm C′ and clutch and cam H to pivot on the fixed knife edge forcing the spring arm E to make contact with the fixed contact F. This completes the electrical circuit and energizes a relay (not illustrated) which actuates the stoker. As the temperature of the firebox rises, pressure on arm C decreases and arms C and E pull away from the contact F. The power circuit, however, is maintained by means of a jumper in the relay until pressure on arm C is reduced to the point where C pulls away from spring arm E. The distance between E and F is, therefore, the differential on which the control operates and is adjustable.

In case of a heat run, the relay is energized independently of the hold-fire control and as the firebox temperature rises, the spring S maintains tension on the moving arm C and draws it back toward stop G. The clutch cam H comes into contact with the stop and due to the shape of the cam, is tilted into position to release moving

arm C from the second or rear notch and permits it to drop into the first or front notch as the cam H passes under the stop G. This occurs at about 900 F with this particular control. When the moving arm C reaches stop G, the cam has righted itself due to the upward pull of a spring T which connects arm C and cam H, locking arm C into the first notch. After the heat run is over, the firebox cools and the pressure on arm C by the heat responsive element forces arm C toward contact F where contact is eventually made as in the normal hold-fire operation. However, being on the first notch instead of the second on cam H, contact C-E-F is made at a temperature higher (100 F in this particular instrument) than normal. After the operating run with clutch set, the circuit is broken in the usual manner and as

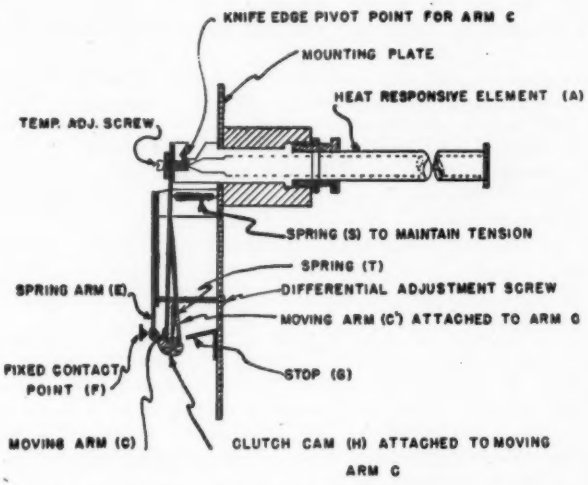
KNIFE EDGE PIVOT POINT FOR ARM C
MOUNTING PLATE
HEAT RESPONSIVE ELEMENT (A)
TEMP. ADJ. SCREW
SPRING (S) TO MAINTAIN TENSION
SPRING (T)
SPRING ARM (E)
DIFFERENTIAL ADJUSTMENT SCREW
MOVING ARM (C') ATTACHED TO ARM C
FIXED CONTACT POINT (F)
STOP (G)
MOVING ARM (C)
CLUTCH CAM (H) ATTACHED TO MOVING ARM C

FIG. 1. DIAGRAMMATIC SKETCH OF FIREBOX TEMPERATURE ACTUATED HOLD-FIRE CONTROL WITH CLUTCH DEVICE TO BRIDGE TRANSITION FROM HEAT RUN TO HOLD-FIRE OPERATION

arm C pulls away from E the rear of the cam H contacts stop G. This trips the cam so that arm C drops back into the second or normal hold-fire operating notch. The control then operates as a normal temperature actuated control until another heat demand occurs.

TEST EQUIPMENT AND PROCEDURE

Four types of hold-fire controls have been tested. These include a number of time lapse devices, a stack temperature control, standard firebox temperature controls, and the firebox temperature control having the clutch device shown diagrammatically in Fig. 1.

Operation has been studied under a variety of conditions including extended hold-fire periods of a week or more in duration, hold-fire periods immediately

following a short operating period sufficient to raise the firebox temperature to about 1000 F (a type of operation commonly encountered during summer service water demand) and hold-fire operation following a prolonged firing period of 15 min or more (mild weather type operation).

One series of tests was conducted using a steam boiler, 530 sq ft EDR, connected to a unit heater, rated 650 sq ft EDR and to an indirect service water heater. The boiler was fired with a 20 lb per hour clinkering type stoker modified to incorporate the preoxidation principle ‡ for control of coke tree formation [3] which produces a relatively flat bed and burns the coal substantially as fast as fed. The coals used in these tests included Pittsburgh seam 1 in. x 0, Upper Freeport seam ¾ in. x 0 and Lower Kittanning seam ¾ in. x 0. All the data herein reproduced are for the Lower Kittanning seam coal, although the results were substantially the same with all the coals tested.

Another series of tests was conducted using a gravity warm air furnace of 81,000 Btu per hour rating operated with air ducts open. The furnace was fired with an ash removal stoker set at a 17 lb per hour feed rate. This unit also incorporated the preoxidation principle. Only data for tests on the Lower Kittanning seam, ¾ in. x 0, coal are reproduced for this series of tests, although Pittsburgh seam, 1 in. x 0, was also tested and gave substantially the same results.

In both furnace and boiler tests, the maximum draft was set at 0.04 in. water gage by means of an automatic draft control. During the hold-fire periods, however, much lower drafts were experienced because stack temperature was relatively low.

Firebox temperatures in the boiler were determined by means of a calibrated unshielded thermocouple inserted through the fire door a distance of 6 in. at a height of 15 in. from the hearth. In the furnace test the thermocouple was inserted through the fire door a distance of 5 in. at a height of 18 in. from the grate. Temperatures were recorded by means of a strip chart recording instrument and also by means of an indicating pyrometer connected in parallel with the recorder.

EXPERIMENTAL RESULTS

The experimental results are shown in Figs. 2 to 7 inclusive for extended hold-fire, socalled summer type operation, and mild weather operation in both furnace and boiler. The plots shown are traced directly from strip chart records except in the case of Fig. 5, where a study was being made of the temperatures of cut-on, cut-off and the extent of overshooting during summer type operation. In these cases, the data are plotted from indicating pyrometer readings taken frequently enough to permit accurate reproduction of the temperature cycles over the periods involved.

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‡ Preoxidation Principle—A portion of the total air required for combustion is introduced near the base of the stoker retort and is mixed with the coal by means of a vertical coal elevating screw. As the aerated coal passes into the heated section of the retort, the oxygen in the air oxidizes the surface of the coal and impedes agglomeration of the particles, thus producing a more free burning coke and eliminating coke tree formation.

DISCUSSION OF RESULTS

Extended Hold-fire

Furnace Tests. Using the time lapse control and operating on a 30 min cycle, outfires resulted in less than a full day when operating at 1.0 and 1.2 lb per hour feed rates. The data are reproduced in Fig. 2 only for the latter where fire was maintained for nearly a full 24 hours. Fire could probably have been maintained indefinitely on this cycle at an operating rate of about 1.5 to 1.7 lb per hour, but such tests were not completed because these rates

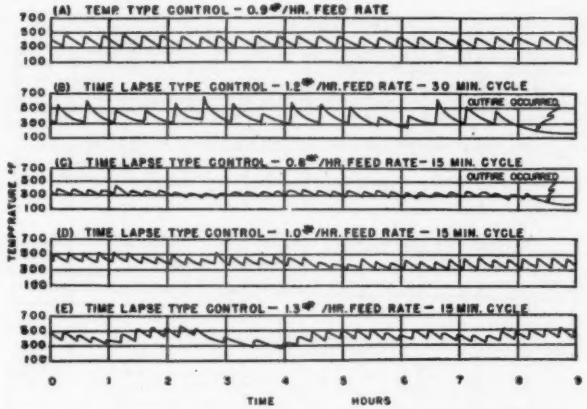
(A) TEMP TYPE CONTROL – 0.9 #/HR. FEED RATE
(B) TIME LAPSE TYPE CONTROL – 1.2 #/HR. FEED RATE – 30 MIN. CYCLE
OUTFIRE OCCURRED
(C) TIME LAPSE TYPE CONTROL – 0.8 #/HR. FEED RATE – 15 MIN. CYCLE
OUTFIRE OCCURRED
(D) TIME LAPSE TYPE CONTROL – 1.0 #/HR. FEED RATE – 15 MIN. CYCLE
(E) TIME LAPSE TYPE CONTROL – 1.3 #/HR. FEED RATE – 15 MIN. CYCLE
TEMPERATURE °F
700
500
300
100
0 1 2 3 4 5 6 7 8 9
TIME HOURS

FIG. 2. EXTENDED HOLD-FIRE TESTS IN GRAVITY WARM AIR FURNACE SHOWING FIREBOX TEMPERATURES USING TWO TYPES OF HOLD-FIRE CONTROLS

are much higher than desirable. When operating on a 15 min cycle, the fire was held for a day at an 0.8 lb per hour rate, but on the second day an outfire resulted as shown in Fig. 2. Satisfactory operation was, however, obtained at 1.0 and 1.3 lb per hour feed rates.

Using the temperature actuated control, satisfactory hold-fire operation was maintained at a 0.9 lb per hour feed rate. A major reduction in the cut-on temperature did not, however, result in an expected decrease in fuel consumption, and this suggests that the 0.9 lb per hour rate is approximately equal to the hold-fire rate necessary to meet the heat losses of this particular installation.

Boiler Tests. As shown in Fig. 3, the fire held satisfactorily with the time lapse control on the 30 min cycle at a rate of 1 lb per hour. Similar results not reproduced were obtained on the 15 min cycle at the 1 lb per hour rate, but on the 60 min cycle at this rate, an outfire resulted in a few hours. At rates of 0.7 lb per hour outfires resulted on the 30 and 60 min cycles but no test has been completed on the 15 min cycle. Using the tem-

perature actuated control, hold-fire rates as low as 0.4 lb per hour were maintained for over two weeks with no apparent difficulty. Fig. 3 shows two sets of operating data, one with a short 60 F differential between cut-on and cut-off and the other with a 140 F differential. The marked difference between the operating characteristics of the time lapse and temperature type controls is apparent.

From these data on furnace and boiler tests it would appear that for extended hold-fire operation where no previous reserve of fuel is available to assist in maintaining fire, the time lapse control may give reasonably satisfactory performance when set on a short operating cycle. It appears to be a characteristic of this type of control that the firebox temperature follows

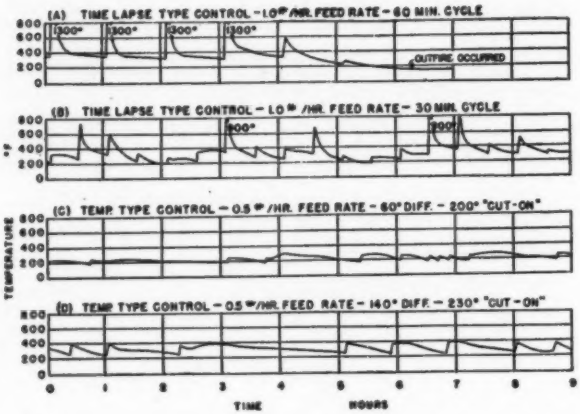

FIG. 3. EXTENDED HOLD-FIRE TESTS IN A STEAM BOILER SHOWING FIREBOX TEMPERATURES USING TWO TYPES OF HOLD-FIRE CONTROLS

a flattened sine wave, the period and amplitude of which increase as the length of the cycle increases. This cyclic rise and fall in the firebox temperature at the time of cut-on results in outfires many times when the average firebox temperature over the period is more than sufficient to expect satisfactory hold-fire operation and high enough in many cases to produce overheating.

With the temperature actuated control, somewhat better results are obtained because no sine wave operation is possible. If the cut-on temperature is set slightly above the firebox temperature necessary to maintain ignition with the particular coal, stoker and furnace or boiler, the fire will hold indefinitely.

Comparing the data for tests with the furnace and with the boiler, it will be noted that the permissible hold-fire rate setting is appreciably lower for the latter. It will also be noted that the general shape of the operating curves for the furnace and boiler tests are quite different when using the temperature actuated control. Both effects are due to the different rates of heat loss from

the fuel bed in the two installations. In the furnace, the radiation and convection losses, which must be met by feeding and burning fuel, are greater than the radiation losses in the boiler. In the latter equipment, operation of the stoker is required principally to maintain ignition of the fuel and to replenish the small quantity of coal burned in maintaining ignition, while in the former, operation is required principally to maintain a fuel bed sufficient to meet the heat losses. The differences between furnace and boiler operation would be expected to be less when using a forced air system because convection

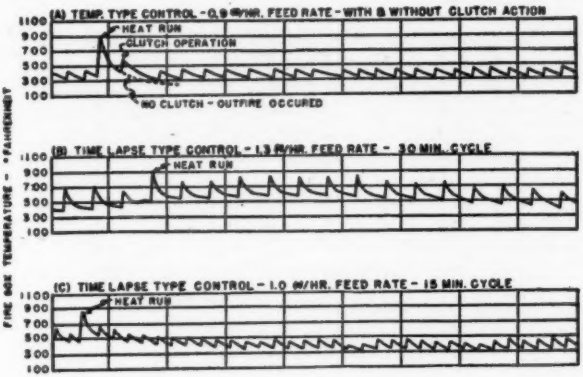
(A) TEMP. TYPE CONTROL - 0.9 #/HR. FEED RATE - WITH & WITHOUT CLUTCH ACTION
HEAT RUN
CLUTCH OPERATION
NO CLUTCH - OUTFIRE OCCURED
(B) TIME LAPSE TYPE CONTROL - 1.3 #/HR. FEED RATE - 30 MIN. CYCLE
HEAT RUN
(C) TIME LAPSE TYPE CONTROL - 1.0 #/HR. FEED RATE - 15 MIN. CYCLE
HEAT RUN
FIRE BOX TEMPERATURE - °FAHRENHEIT
1100
900
700
500
300
100

FIG. 4. SUMMER TYPE HOLD-FIRE OPERATION IN A GRAVITY WARM AIR FURNACE AT VARIOUS RATES SHOWING FIREBOX TEMPERATURE CONDITIONS IMMEDIATELY FOLLOWING A SHORT OPERATING RUN

losses would be somewhat reduced in this type of equipment. Tests have not yet been made to substantiate this belief.

Summer Type Operation

Furnace Tests. Test results on summer type operation using the time lapse control on a 15 min cycle gave satisfactory performance at a 1.0 lb per hour hold-fire rate, as shown in Fig. 4. On the 30 min cycle, however, an outfire occurred about 10 hours after the short operating run, at the first low point in the wave cycle. Although the data shown in Fig. 4 do not include the outfire, its occurrence indicates that this hold-fire rate is too low to maintain fire under these operating conditions. It is apparent from both of these time lapse tests that considerable overheating of the home would occur from operating at the hold-fire rates tested. Since furnaces are not generally used for service water production during the summer, this particular test is only of significance as indicating what may be expected during mild weather after even the shortest of operating periods.

The simple temperature actuated control not equipped with clutch action to give the fire a boost during the cooling period between peak temperature

and normal hold-fire temperature gave results worse than did the time lapse control. The fire was out by the time the control called for operation. In this particular test, the outfire occurred 40 min after the heat run and at least 20 min before the control called for operation. Obviously the outfire would have been prevented if the cut-on temperature had been raised 60-80 F, but this would have resulted in a hold-fire rate appreciably higher than 0.9 lb per hour. Although the time lapse control was not tested on the 60 min cycle, it is obvious from the foregoing data that an outfire would have

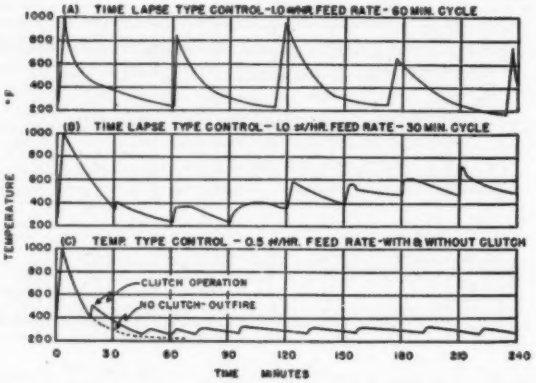

FIG. 5. SUMMER TYPE HOLD-FIRE OPERATION IN A STEAM BOILER AT VARIOUS RATES SHOWING FIREBOX TEMPERATURE CONDITIONS IMMEDIATELY FOLLOWING A SHORT OPERATING RUN

resulted on this cycle at the 0.9 or 1.0 lb per hour rate before the control called for operation.

As may be seen from the data shown in Fig. 4, the overheating difficulties inherent in the time lapse method of hold-fire control are eliminated to a large extent by using the clutch device in conjunction with the temperature actuated control. Not only is it possible to operate on a lower hold-fire setting as has been shown for extended hold-fire operations, but after a short operating period the firebox temperature returns rapidly to normal hold-fire temperature. The extra boost given to the fire about 100 to 150 F above normal hold-fire temperature adds very little to the average firebox temperature, yet is adequate to prevent the outfire frequently experienced with the simple temperature actuated control without clutch device.

Boiler Tests. In the tests reproduced in Fig. 5 the data were taken from indicating pyrometer reading of cut-on, cut-off and maximum temperatures following each operating period in order to study more carefully the temperature variations occurring. Only four hours of tests are plotted although in all cases the tests were continued for a day or more and the fires were held satisfactorily.

These data show the marked difference in firebox temperatures when operating on the different time cycles and on the firebox temperature type control. It is especially interesting to note that on the 60 min cycle the firebox temperature dropped to 185 F at the end of the third hold-fire cycle, which is at least 15 F below what is considered the safe minimum operating temperature. Similarly on the 30 min cycle relatively poor response to the *on* periods was obtained on the first and second cycles until a small reserve of fuel had accumulated which was ignited on the third cycle and continued to burn during the subsequent *off* period. The amount of coal fed and burned during the time

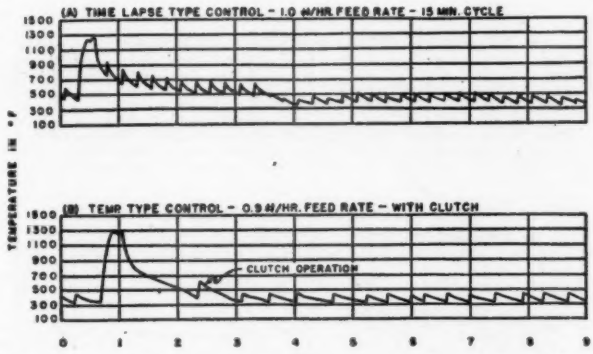

FIG. 6. FIREBOX TEMPERATURE CONDITIONS IN A GRAVITY WARM AIR FURNACE IMMEDIATELY FOLLOWING A 15 MIN HEAT RUN, USING TWO TYPES OF HOLD-FIRE CONTROLS

control tests was at least twice that necessary for normal hold-fire yet a lower setting was not possible because of the conditions discussed.

Mild Weather Operation

Furnace Tests. Fig. 6 shows the strip chart records for tests with the time lapse type control operating on a 15 min cycle at a hold-fire rate of 1 lb per hour, and for the temperature actuated control with clutch action operated at a 0.9 lb per hour hold-fire rate. The marked difference in firebox temperatures is at once apparent. For the time cycle, fire was almost lost about 3½ hours after the 15 min heat run due to the fact that the high firebox temperatures caused the fuel to burn faster than it was being fed. With the temperature actuated control, however, this control did not call for operation of the stoker for 70 min while the firebox cooled to 400 F where a short operating period on clutch occurred, sufficient to hold the fire until normal hold-fire was attained some 50 min later. Had the clutch not caused a short run at this time, the fire would have been lost. In this particular installation with the particular coal tested, it has been found that the fire will lose ignition about 85 to 90 min after a heat run.

Boiler Tests. Similar data for the tests with both types of controls in the boiler are shown in Fig. 7. For the time lapse type control, only the data for the 30 min cycle at the 1 lb per hour feed rate are reproduced, although a test on the 15 min cycle was completed and resulted in satisfactorily maintaining the fire for at least a full day of operation. A test using the 60 min cycle at the 1 lb per hour feed rate gave satisfactory operation until the reserve of fuel burned out, after which an outfire resulted as with extended hold-fire on this cycle.

For the temperature actuated control, tests were made with and without clutch action. The latter resulted in an outfire which occurred approximately

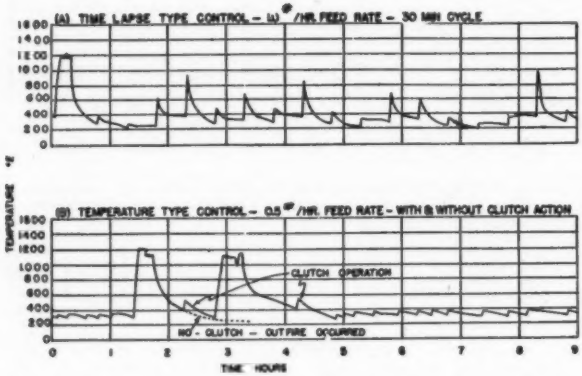

FIG. 7. FIREBOX TEMPERATURE CONDITIONS IN A STEAM BOILER IMMEDIATELY FOLLOWING A 15 MIN HEAT RUN, USING TWO TYPES OF HOLD-FIRE CONTROLS

70 min after completion of the heat run at a firebox temperature of 360 F, or some 80 F above normal hold-fire cut-on temperature.

To demonstrate the effectiveness of the clutch action in maintaining fire under adverse conditions, two 15 min heat runs were made in the test reproduced in Fig. 7. The second 15 min run was started as soon as the firebox temperature had dropped to that of normal hold-fire. As may be seen, the fire responded well and held satisfactorily on subsequent hold-fire.

The marked difference between the firebox temperatures when using the two different types of controls is immediately apparent. With the time control, the temperatures reached 600 F or higher on six of the subsequent hold-fire operating cycles and during a substantial portion of the nine hours following the heat run the firebox temperature was above 400 F. In comparison, the temperature control permitted the firebox to cool to 400 F in 60 min after the second 15 min heat run and had reached normal hold-fire of 280 F cut-on, 320 F cut-off temperature in 100 min.

Similar results have been obtained after heat demand periods of longer than 15 min except that the reserve of fuel thus built up was greater and the

time lapse type of control tends to hold the firebox temperatures higher for longer periods while the temperature actuated control attains normal hold-fire a little more slowly. The greater the reserve of fuel in the firebox the greater the difference in the extent of overheating between operation with the two types of controls.

CONCLUSIONS

The results of tests with a number of different types of hold-fire controls under a variety of operating conditions in different types of heating equipment have demonstrated that hold-fire controls actuated by changes in firebox temperatures are intrinsically capable of more reliable and efficient operation with less danger of overheating the home than are the time lapse type or stack temperature type controls.

The standard type of firebox temperature actuated control fails, however, to render the effective performance possible, because during the transition period from heat demand operation to hold-fire operation, the fuel bed will frequently cool faster than the surrounding refractories or castings and loss of ignition will occur under certain conditions before the temperature control calls for operation, unless the cut-on temperature and, therefore, the hold-fire rate is appreciably higher than required for normal hold-fire operation.

A new firebox temperature actuated control with a clutch device to bridge the transition period from heat run to hold-fire operation has been found to overcome the deficiency of the standard control. This control permits satisfactory, safe and efficient operation at hold-fire rates appreciably lower than any previous control tested. The use of controls of this type may be expected to improve the reliability and efficiency of bituminous coal stoker performance during mild weather and summer service water operation and to increase consumer satisfaction with bituminous coal stokers.

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3. Control of Coke Tree Formation in Domestic Underfeed Stoker, by C. C. Wright and T. S. Spicer, *Mineral Industries Experiment Station Technical Paper No. 79* (1942). *Transactions, American Institute of Mining and Metallurgical Engineers, 149,* 270-282 (1942).

DISCUSSION

R. A. SHERMAN, Columbus, Ohio: I quite agree with Doctor Wright that the time-lapse control is wrong in principle, although it is so exceedingly simple that it has worked very satisfactorily with many stokers, and with many coals. The trouble is, of course, the over-run of temperature in the house and also the amount of coal which is used. It has been general experience that stokers are somewhat inefficient in mild weather operation, as compared to other fuels in which you do not need a

hold-fire control. I did not understand quite clearly just how much reduction in hourly consumption of coal you could get with firebox temperature control and I would like to know what Doctor Wright has decided as a maximum that should be allowed. We have been interested in some standard by which to judge the performance of stokers and of stoker coals.

R. A. BAKER, Syracuse, N. Y.: Is there a satisfactory type of temperature operated hold-fire control on the market?

W. H. CARRIER, Syracuse, N. Y.: I have had some experience with a time-lapse type of control on an anthracite type of stoker on which the original adjustment has not needed any change. It has been in continuous operation over a six-year period and is satisfactory in every way.

W. A. DANIELSON, Memphis, Tenn.: I have a time-lapse type of control that has worked very satisfactorily in a stoker installation burning anthracite on a hot water installation equipped for summer-winter hot water supply. The boiler thermostat maintained a fire with a water temperature just below 100 F.

AUTHORS' CLOSURE: Referring to Mr. Sherman's comments for a home installation the hold-fire rate to take care of the heat loss from the service water tank should be about ½ lb of coal per hour, whereas the lowest rate obtainable by means of the time-lapse control was 1.3 lb per hour, which was about what would be normal for winter operation. The temperature type of control has operated with the same type of coal at one-half pound per hour on a clinkering stoker and at a rate of 0.36 lb per hour with an ash removal type of stoker. While samples of the temperature type of control are in field operation durability has not yet been determined by operation for a sufficiently long period. Three control manufacturers are working on the production of types of control employing the principle described.

Referring to the remarks of General Danielson and Mr. Carrier; while operation with anthracite would be satisfactory with time lapse controls, operation with most bituminous coals and bituminous coal stokers would be improved by the use of a temperature type of control.

1303

No. 1303

THE INFLUENCE OF AN INTERIOR COATING OF ALUMINUM PAPER ON INTERNAL THERMAL CONDITIONS AND HEAT ECONOMY

By R. J. Lorenzi,* L. P. Herrington,** and C.-E. A. Winslow,†
New Haven, Conn.

Object and Nature of Experiments

IN AN earlier contribution[1] there was reported a comparison of internal thermal conditions and heat economy in two experimental rooms, one of brick and one of ordinary frame construction. The present paper relates to a study made of the brick room, used in the earlier experiment, after covering the lower surface of its ceiling and the interior surface of its walls with a commercial aluminum product consisting of aluminum foil mounted on one side of kraft paper. The thermocouples on the aluminum surface were separated from that surface by cellulose tape to prevent electrical transfer to the metallized surface. The thermocouples were held in place by an additional layer of cellulose tape placed over the junction. In all other respects, the experiments were identical with those made previously with the plaster walls of the same room. It is, therefore, unnecessary to report the structural and operating details given fully in our earlier contribution.

The room was 15 x 12 x 8 ft high. Its north, west and south walls faced exterior cooled shell spaces, with two windows on the long west wall and one on the short south wall. The east wall faced an interior space.

Comparable experiments were made under three conditions:

1. With a continuous external temperature of 20 F; interior of room maintained at 70 F for a preliminary stable period; heating lamps turned off for 8 hours, to repre-

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** Associate Director, John B. Pierce Laboratory of Hygiene.
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[1] The Influence of Heat Capacity of Walls on Interior Thermal Conditions and Heat Economy, by C.-E. A. Winslow, L. P. Herrington and R. J. Lorenzi. (A.S.H.V.E. Transactions, Vol. 51, 1945, p. 197.)

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Montreal, Que., Canada, June, 1946.

sent a period of night cooling; after which the interior of the room was rapidly heated up to normal again. (Experiments B-II and B-Al-II, without and with aluminum, respectively.)

2. The same experiment, except that during the period of night cooling the lower sash of the south window of the room was raised 8 in. and the upper sash of the

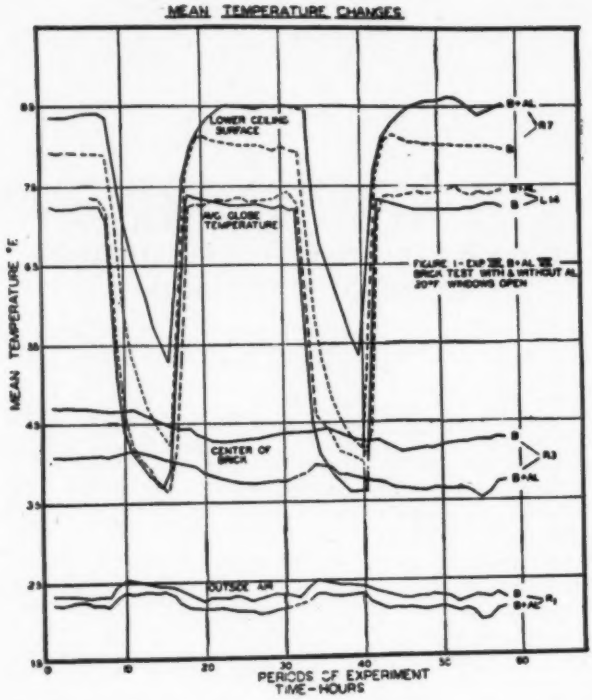

FIG. 1. MEAN TEMPERATURE CHANGES DURING TESTS CONDUCTED WITH CONTINUOUS 20 F EXTERNAL TEMPERATURE

northerly window on the west wall was lowered 8 in. (Experiments B-VII and B-Al-VII.)

3. The same as described under (A), except that instead of a constant exterior temperature of 20 F, the external temperature was cycled between 10 F and 30 F, to represent a diurnal cycle, the period of night cooling coming at the low point of the external cycle. (Experiments B-III and B-Al-III.)

The results of B-VII and B-Al-VII and of B-III and B-Al-III, are shown graphically in Figs. 1 and 2. In each case, the night cooling cycle was repeated twice.

Fig. 1 (for the experiments with constant 20 F temperature outside and night cooling with windows open) shows that the external air was nearly the same in the two experiments compared, although a little over 1 deg lower in B-Al-VII. The mean globe temperature during the stable phase was about 1.5 deg higher in B-Al-VII. These were, of course, differences due to the experimental set-up.

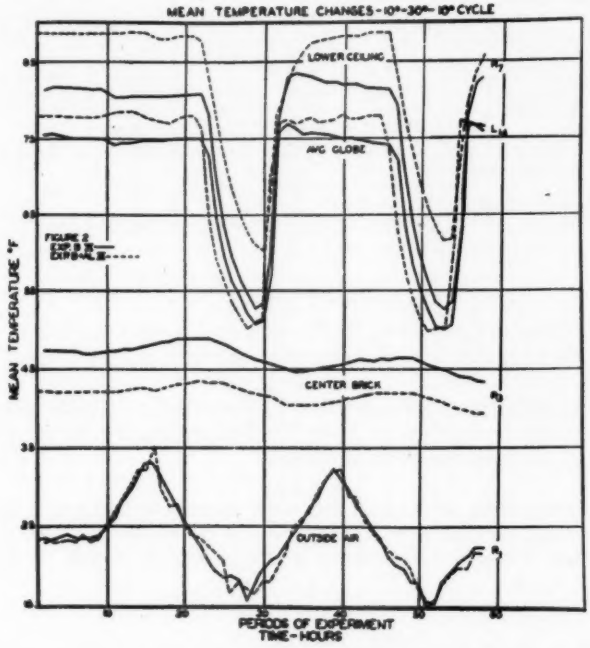

FIG. 2. MEAN TEMPERATURE CHANGES DURING TESTS CONDUCTED WITH 10 F—30 F—10 F TEMPERATURE CYCLES

The very much higher temperatures for B-Al-VII at the lower ceiling and the much lower temperature at the center of the brick wall were, on the other hand, characteristic results of the use of the aluminum insulation. These differences will be discussed in detail in later paragraphs.

Similar results for the experiments with cycling outdoor temperature are shown in Fig. 2 for Experiments B-III and B-Al-III. In this case, the mean globe temperature during the stable phases was about 3 deg higher in B-Al-III than in B-III. This was due to experimental procedure. The higher ceiling temperature and the lower wall temperature for the aluminum experiments are similar to those reviewed in the foregoing.

TABLE 1—MEAN STABLE PHASE TEMPERATURES OF EXTERNAL ENVIRONMENT

	PRELIMINARY STABLE PHASE					
	BRICK			BRICK—ALUMINUM		
	Exp. II	Exp. III	Exp. VII	Exp. II	Exp. III	Exp. VII
Mean Air Outside 3 Exterior Walls	21.9	20.6	21.9	20.6	19.8	20.3
Mean of Outside Surface, 3 Exterior Walls	29.7	29.3	29.3	26.7	27.0	26.2
Outside Surface, Windows	34.6	34.2	34.0	33.3	34.8	33.1
Air above Ceiling	49.2	49.4	50.1	48.2	51.1	48.6
Upper Ceiling Surface	50.7	51.0	51.8	51.0	53.0	50.8
Air below Floor	49.5	49.4	51.8	48.8	50.4	51.6
Lower Floor Surface	50.8	51.0	53.0	50.9	51.7	53.2
Air Outside East Interior Wall	60.8	63.5	66.2	62.3	63.1	64.6
Outside Surface, East Interior Wall	59.0	61.3	64.4	61.3	63.4	65.8

COMPARISON OF TEMPERATURE CONDITIONS MAINTAINED WITH AND WITHOUT ALUMINUM INSULATION DURING STABLE PHASE

The mean stable phase temperatures of the outside environment for the six experiments are summarized in Table 1. Similar data for the interior temperatures of the room are given in Table 2.

It is apparent from these tables that some variation between the six experiments did occur at some points. However, these differences which were due to limitations in experimental control can, in reasonable degree, be eliminated

TABLE 2—MEAN STABLE PHASE TEMPERATURES OF INTERIOR OF ROOM

	PRELIMINARY STABLE PHASE					
	BRICK			BRICK—ALUMINUM		
	Exp. II	Exp. III	Exp. VII	Exp. II	Exp. III	Exp. VII
Inside Surface, Windows	60.0	60.4	58.8	60.2	61.4	58.5
Mean of Inside Surface of 3 Exterior Walls	61.8	61.5	60.7	56.7	58.6	55.0
Upper Floor Surface	72.6	73.5	71.0	75.1	76.1	72.5
Inside Surface, East Interior Wall	73.4	73.7	71.9	72.0	73.2	70.0
Mean Room Air	74.1	73.7	71.6	72.9	74.5	70.0
Mean Globe	75.6	75.2	72.3	76.1	77.8	73.0
Lower Ceiling Surface	80.9	81.4	79.1	86.4	88.5	83.7

by averaging the three experiments for each structure, although the aluminum series is slightly weighted by higher figures for B-Al-VII.

A comparison of mean results is made in Table 3, with the data arranged according to the flow of heat through the five surfaces involved.

Table 3 brings out the true characteristics of the aluminum insulation. The outside air in the aluminum experiments averaged 1.3 deg lower than in the brick series. This was due to basic control conditions, independent of the insulating effect. But the fact that the outside surface of the exterior walls was 2.8 deg lower and their inside surfaces 4.5 deg lower is highly significant.

TABLE 3—STABLE PHASE TEMPERATURES RECORDED AT VARIOUS POINTS (MEAN OF B-II, B-III AND B-VII, COMPARED WITH MEAN OF B-AL-II, B-AL-III AND B-AL-VII)

	BRICK	ALUMINUM	DEVIATION OF ALUMINUM
Outside Air	21.5	20.2	−1.3
Outside Wall Surface	29.4	26.6	−2.8
Inside Wall Surface	61.3	56.8	−4.5
Mean Air	73.1	72.5	−0.6
Mean Globe	74.4	75.6	+1.2
Outside Window Surface	34.3	33.7	−0.6
Inside Window Surface	60.0	59.7	−0.3
Air above Ceiling	49.6	49.3	−0.3
Upper Ceiling Surface	51.2	51.6	+0.4
Lower Ceiling Surface	80.5	86.2	+5.7
Air below Floor	50.2	50.3	+0.1
Lower Floor Surface	51.6	51.9	+0.3
Upper Floor Surface	72.4	74.6	+2.2
Air Outside East Wall	63.5	63.3	−0.2
Outside Surface, East Wall	61.6	63.5	+1.9
Inside Surface, East Wall	73.0	71.7	−1.3

There were no significant differences in the window temperatures, or in the temperatures above the ceiling, or below the floor, or in the space outside the interior east wall.

The lower ceiling surface was 5.7 deg higher with the aluminum wall and the upper floor surface (due to reflection from the ceiling) was 2.2 deg higher. The higher ceiling surface temperature was due to the stratification of warm air at the ceiling caused by a reduced film coefficient. So far as the net influence of radiation and convection was concerned, the mean air temperature of the aluminum room was 0.6 deg lower than that of the brick room and its mean globe temperature was 1.2 deg higher.

TEMPERATURES ATTAINED DURING COOLING PHASE WITH AND WITHOUT ALUMINUM INSULATION

The minimum temperatures attained during the cooling phase are indicated in Table 4.

TABLE 4—MINIMUM TEMPERATURES REACHED INSIDE ROOM DURING FIRST COOLING PHASE

	EXP. II		EXP. III		EXP. VII	
	B-II	B-AL-II	B-III	B-AL-III	B-VII	B-AL-VII
Inside Window Surface	42.9	42.6	41.3	41.0	34.2	32.6
Inside Surface, West Wall	50.7	46.1	50.4	47.0	43.3	40.0
Inside Surface, South Wall	50.6	46.2	50.4	47.4	43.0	40.4
Inside Surface, North Wall	50.2	46.4	50.2	47.6	42.6	40.3
Upper Floor Surface	51.9	51.8	52.1	51.4	39.2	40.9
Inside Surface, East Wall	52.7	52.8	52.4	52.9	40.8	41.2
Mean Air	51.6	50.9	50.8	50.2	35.8	36.6
Mean Globe	51.6	50.8	50.9	50.3	36.9	36.7
Lower Ceiling Surface	52.7	60.8	52.9	60.4	42.3	52.8

It will be noted that results for Experiments B-II and B-III were practically identical, and that results for B-Al-II and B-Al-III were practically identical with each other, although wall surfaces were 2.3 deg—4.6 deg lower than without the aluminum insulation and the lower ceiling, 7.5 deg—10.5 deg higher.

Experiments B-VII and B-Al-VII, with the windows open, showed, of course, lower minimum temperatures than B-II, B-III, B-Al-II and B-Al-III because of the open windows. The processes of cooling and reheating were

TABLE 5—RATES OF COOLING AND REHEATING OF BRICK AND ALUMINUM SURFACED ROOMS

(Fall of globe temperature in first 2 hours of cooling)

	EXPERIMENT		
	II	III	VII
B	11.7	10.1	26.3
B-Al	17.3	17.1	26.3

(Rise in globe temperature in first 30 min of reheating)

	EXPERIMENT		
	II	III	VII
B	20.0	18.8	26.0
B-Al	27.0	27.8	32.5

much more rapid in the case of the aluminum wall as indicated in Table 5 (with the exception of the cooling phase in the open window tests).

STABLE PHASE TEMPERATURES ON A GRADIENT THROUGH NORTH WALL WITH AND WITHOUT ALUMINUM INSULATION

In addition to the data reported, 6 thermocouple records were obtained at the following points, giving a gradient through the north wall; inside air 4 in. from the center of the wall; inside surface at center of the wall; contact between interior plaster and brick; center of the brick wall; exterior surface of the brick wall; air 4 in. out from the exterior brick surface.

Mean temperatures at these six points for the six experiments, during the preliminary stable period, are summarized in Table 6.

In general, the results for the brick series check each other as do the results for the brick-aluminum series. Experiments B-VIII and B-Al-VII both had low inside air temperatures which were reflected in low temperatures at the

TABLE 6—MEAN TEMPERATURES AT VARIOUS POINTS ON A GRADIENT THROUGH NORTH WALL

	PRELIMINARY STABLE PHASE							
	Brick				Brick-Aluminum			
	Experiment			Avg.	Experiment			Avg.
	II	III	VII		II	III	VII	
Outside Air	22.5	23.0	23.2	22.9	21.6	22.8	22.4	22.3
Outside Surface	33.0	33.8	33.4	33.4	29.6	30.6	29.6	29.9
Center of Brick	46.2	47.2	46.8	46.7	41.0	42.0	40.8	41.3
Brick-Plaster Contact	61.5	62.5	61.3	61.8	53.2	56.1	51.9	53.7
Inside Surface	63.5	64.3	62.5	63.4	58.3	60.8	56.7	58.6
Inside Air	71.8	71.8	69.9	71.2	70.2	73.5	68.5	70.7

inside surface and the brick-plaster contact. In Experiment B-Al-III these three points showed high temperatures as a result of warmer inside air.

The data are sufficiently comparable to warrant averaging within each series. Using this combined average, the brick aluminum wall shows an outside air temperature 0.6 deg lower; outside surface temperature 3.5 deg lower; center of brick temperature 5.4 deg lower; brick-plaster contact temperature 8.1 deg lower; inside surface temperature 4.8 deg lower; and inside air temperature 0.5 deg lower than the uninsulated brick wall. The general influence with the aluminum insulation is apparent.

Since the temperature differentials between brick-plaster contact and inside surface in the aluminum series appeared to be greater than we had anticipated, we were led to suspect that the inside surface temperatures as recorded might be too high, due to reflective effects upon the surface thermocouples. Special

studies reported in a later paragraph confirmed the view that our readings on the aluminum wall surfaces were about 2.2 deg too high. Thus, in Table 6, the inside surfaces should be, respectively, 56.1 deg, 58.6 deg and 54.5 deg for Experiments B-Al-II, B-Al-III, and B-Al-VII and the average should be 56.4 deg.

INFLUENCE OF ALUMINUM INSULATION ON WALL CONDUCTANCE

For the determination of the influence of the aluminum insulation upon wall conductance, a special experiment, B-Al-V, was conducted. In this experiment, the shell space outside the north wall was kept at 15 deg and the other three shell spaces, the spaces above the ceiling and below the floor and the interior of the room itself, were all maintained at about 90 deg. The actual mean values at various points in the room and its environment varied from 85.5 deg on the outside surface of the west wall to 90.4 deg on the inside of the east wall, except that the lower ceiling surface was 96.5 deg. The mean inside air temperature was 89.5 deg, and the mean globe temperature was 90.8 deg. These conditions were essentially identical with those obtained in a similar experiment, B-V, before the aluminum coating was applied.

The temperature differentials between successive points on the gradient through the north wall were as follows for the two experiments:

	WITHOUT ALUMINUM	WITH ALUMINUM
Inside Air—Inside Surface	13.8	16.8
Inside Surface—Brick-Plaster Contact	1.6	7.1
Brick-Plaster Contact—Center of Brick	23.3	20.1
Center of Brick—Outside Surface	20.1	16.6
Outside Surface—Outside Air	14.8	11.0

These results correspond to those obtained (under less extreme conditions) in Experiments B-Al-II, B-Al-III and B-Al-VII. Again, the very great drop between the inside surface and the brick-plaster contact is not reasonable. The surface thermocouple temperatures as recorded on the aluminum walls and ceiling and on the floor were, therefore, checked by thermopile readings with the following results:

	THERMOCOUPLE TEMPERATURES	THERMOPILE TEMPERATURES
North Wall	67.0	91.4
West Wall	87.5	91.7
South Wall	88.0	91.1
East Wall	90.5	91.2
Lower Ceiling	96.5	89.9
Upper Floor	89.0	89.1

On the floor, which is nonreflective, the results checked, and on the lower ceiling the thermocouple reading was 6.6 deg higher than that of the thermopile. In the latter case, the higher thermocouple reading was probably influenced by the accumulation of warm air below the ceiling surface. In the case of the west, south and east walls, on the other hand, the thermocouple temperatures were lower than the thermopile temperatures by 4.2 deg, 3.1 deg and 0.7 deg, respectively. In the case of the cold north wall, the thermocouple figure was 24.4 deg lower than the thermopile temperature. The thermopile temperatures are, of course, significant with respect to the radiant effect of the surfaces. The thermocouple temperatures, on the other hand, should indicate heat loss from the surface to the exterior. The possibility remains, however, that the recorded thermocouple temperatures were somewhat higher than they should be on account of direct radiative effects of other aluminum surfaces on the couple itself. We tested this point by protecting the thermocouple on the north wall by a 12 in. x 12 in. aluminum shield placed 0.5 in. from the walls. The thermopile temperature of the north wall in this case was 74.0 deg. The thermocouple read 54.8 deg without the shield and 52.8 deg with the shield. We, therefore, concluded that the aluminum foil surface temperatures as recorded were about 2 deg too high. A more complete analysis of this question will be offered in a succeeding paragraph.

The major purpose of Experiment B-Al-V was to determine the heat conductance of the aluminum covered wall. The mean heat input into the room for the stable condition was 687 watts, or 2336 Btu per hour. By using GUIDE coefficients for inside air to outside air, we computed a loss of 139 Btu through the ceiling, a gain of 6 Btu through the east wall, a loss of 28 Btu through the windows, a loss of 49 Btu through the floor, a loss of 66 Btu for the west wall and a loss of 13 Btu through the south wall. The balance (2336 − 289) was 2047 Btu passing through the north wall. This gives an air-to-air coefficient of 0.23 Btu per (hr) (sq ft) (F deg) for the aluminum covered north wall, identical with the theoretically assumed coefficient for the west and south walls. This figure compares with a figure of 0.29 previously determined for the same wall without aluminum. In all cases, as noted in an earlier paper, computations were made on the assumption of a total surface area midway between internal and exterior surfaces.

The surface-to-surface coefficient for the aluminum covered wall, determined from the thermocouples as installed, was 0.40 Btu per (hr) (sq ft) (F deg), as compared with 0.44 for the wall before the aluminum insulation was added, which is obviously too great a difference. Knowing the actual flow of heat through the north wall [air-to-air, 17.0 Btu per (hr) (sq ft)], we computed the flow between each of the succeeding gradient points through the wall and found that the individual figures checked the total overall figure reasonably well, except for the flow from inside surface to brick-plaster contact, which was 24.8 Btu per (hr) (sq ft). This confirmed our conclusion that the inside surface temperature was about 2.0 deg too high. By trial and error we found that if the recorded inside surface temperature be considered 2.2 deg too high, the flow between successive points on the gradient is entirely reasonable. On this assumption, the flow from inside air to inside surface was 17.4 Btu per

(hr) (sq ft); that for inside surface to brick-plaster contact was 17.1; and that for brick-plaster contact to center of brick was 16.6 Btu. Using inside surface temperatures 2.2 deg below those recorded, the overall surface-to-surface coefficient with aluminum works out to 0.43 Btu per (hr) (sq ft) (F deg) as compared with 0.44 for the original brick wall. The added obstacle to conductance introduced by the aluminum foil with its paper backing was very low. Its conductance was estimated as 10 Btu per (hr) (sq ft) (F deg).

The effect of the aluminum foil is primarily to reduce inside film coefficients by a reduction in heat transfer to the wall surface due to its reflective capacity. Comparing B-V and B-Al-V, we find that while the air-to-air overall temperature differentials and the surface-to-surface coefficients are practically the same in the two experiments, the air-to-air coefficient is reduced by 20.7 per cent, the actual decrease in required power input being 20.8 per cent, as indicated in the following:

	AIR-TO-AIR COEFFICIENT, BTU PER (HR) (SQ FT) (F)	ACTUAL BTU REQUIRED TO HEAT ROOM
Exp. B-V	0.29	2951
Exp. B-Al-V	0.23	2336
Difference	0.06	615
Per cent reduction	20.7	20.8

DETERMINATION OF THE INSIDE FILM COEFFICIENT

Using the data of Experiment B-Al-V, it was possible to determine an inside film coefficient for the aluminum covered vertical surfaces by means of three

TABLE 7—SUMMARY OF ACTUAL POWER REQUIREMENTS

BRICK STRUCTURE

	20 F		10–30 F	
1. General Conditions				
2. Experiment Number	B-II	B-Al-II	B-III	B-Al-III
3. Hours, Stable Phase	1–10	1–12	1–7	1–9
4. Mean Watts, Stable Phase	2269	1799	2256	1870
5. Mean Inside Air, Stable Phase	74.1	72.9	73.8	74.5
6. Mean Outside Air, Stable Phase	21.9	20.6	20.7	19.8
7. Temperature Differential	52.2	52.3	53.1	54.7
8. Watts per deg F Differential	43.4	34.4	42.5	34.2
9. Hours, Heating Phase and Second Stable Phase	19–44	21–39	30–45	31–44
10. Total Watt Hours, Heating Phase and Second Stable Phase	68867	40090	43728	31570
11. Mean Inside Air Temperature, Heating and Second Stable Phase	74.4	72.5	72.4	73.0
12. Mean Outside Air Temperature, Heating and Second Stable Phase	21.6	21.3	23.8	24.4
13. Temperature Differential	52.8	51.2	48.6	48.6
14. Theoretical Watt Hours, Heating and Second Stable Phase	59580	33464	33048	23270
15. Excess Watts to Satisfy Heat Capacity (10–14)	9287	6626	10680	8300

methods. In Methods I and II the total amount of heat transferred per hour through the three exposed walls (north, west and south) was obtained by deducting the heat lost from all other surfaces from the total stable period heat input.

In Method I, by using the inside surface area of the exposed walls and the proper temperature differentials, it was possible to calculate two conductance values, one on the basis of inside surface to outside surface $U_{s\text{-}s}$, and the other on the basis of inside air to outside surface $U_{a\text{-}s}$. By then employing the following formula,

$$\frac{1}{U_{s\text{-}s}} = \frac{1}{f_1} + \frac{1}{U_{a\text{-}s}},$$

f_1, the inside film coefficient, was found to be 0.73 Btu per (hr) (sq ft) (F deg).

In Method II, two other conductance values were obtained, one on the basis of inside air to outside air $U_{a\text{-}a}$, and the other on the basis of inside surface to outside air $U_{s\text{-}a}$. By using a formula similar to that employed for Method I, f_1 was again determined to be 0.73 Btu per (hr) (sq ft) (F deg).

In Method III, the radiant transfer and the convective transfer were determined separately and then combined. Using the well known Stefan-Boltzmann Equation

$$q_r = A e_{eff} \sigma (T_1^4 - T_2^4),$$

in which

q_r = the net rate of heat radiation,
A = the area,
e_{eff} = the effective emissivity,
σ = Stefan's constant,
T_1 & T_2 = the absolute temperatures, q_r was determined to be 0.702 Btu per (hr) (sq ft), or 0.028 Btu per (hr) (sq ft) (F deg)

TABLE 8—ESTIMATE OF SAVING BY NIGHT COOLING

BRICK STRUCTURE

	B-II	B-Al-II	B-III	B-Al-III
1. Experiment Number and General Conditions	20 F	20 F	10–30 F	10–30 F
2. 24 Hour Differential	50 F	50 F	50 F	50 F
3. Total Watt Hours, without Night Cooling	52080	41280	51000	41040
4. Temperature Differential Stable Phase	50 F	50 F	46.5 F	46.5 F
5. Total Watt Hours, Stable Phase (16 Hours)	34720	27520	31620	25445
6. Excess Watt Hours, Cooling Phase	990	720	1040	787
7. Excess Watt Hours, Heating Phase	9287	6626	10680	8300
8. Sum of 5, 6 and 7	44997	34866	43340	34532
9. Per Cent Saving by Night Cooling	13.6	15.5	15.0	15.9

The film coefficient for convective transfer was then determined from data of Wilkes and Peterson,[2] and was found to be 0.755 Btu per (hr) (sq ft) (F deg), which gave a combined film coefficient of 0.78 Btu per (hr) (sq ft) (F deg).

The average of the three methods gave a combined film coefficient of 0.75, which checks nicely with the present GUIDE value of 0.74 Btu per (hr) (sq ft) (F deg), for a surface of 0.05 emissivity.

INFLUENCE OF ALUMINUM INSULATION ON HEATING ECONOMY

On the basis of Experiments B-Al-II and B-Al-III, a computation was made of the economy effected by night cooling as compared with the results previously reported for Experiments B-II and B-III. Table 7 and 8 are comparable to those previously presented and an explanation of the method by which the computations were made will be found in an earlier communication.[3]

It will be noted that the addition of the aluminum coating, in addition to obtaining the 21 per cent reduction in heating requirements during a stable state, also increased the additional saving due to night cooling. For comparable conditions, inside and out, the actual percentage heating load would be as follows, with and without aluminum:

	WITHOUT ALUMINUM		WITH ALUMINUM	
	B-II	B-III	B-Al-II	B-Al-III
Without Night Cooling	100	100	79	79
With Night Cooling	86	85	67	66

INFLUENCE OF ALUMINUM INSULATION ON SUMMER COMFORT CONDITIONS

As in earlier experiments, without aluminum insulation, the authors conducted one experiment (B-Al-VI) covering a series of three diurnal cycles, in which no heat was supplied to the room, but in which the temperatures of the shell spaces outside the three exterior walls were varied between 65 F and 95 F. The space below the floor was kept at about 50 F, and the interior of the room, the space above the ceiling and that outside the interior east wall were allowed to drift. The results were essentially the same as those presented in our earlier paper for the brick wall without insulation.

The brick wall provided great protection against the daytime heat. While the outside air ranged from 65.0 F to 93.2 F in Experiment B-VI, and from 64.1 F to 94.1 F in Experiment B-Al-VI, the inside air varied only between 72.5 F and 73.8 F in Experiment B-VI and from 71.4 F to 73.2 F in Experi-

[2] Radiation and Convection from Surfaces in Various Positions. G. B. Wilkes and C. M. F. Peterson. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 513.)
[3] See Footnote 1.

ment B-Al-VI. The aluminum insulated wall showed a maximum outside air temperature 0.9 deg higher than that for the uninsulated room, and a maximum inside air temperature 0.6 deg lower than that for the uninsulated room, a very minor advantage in favor of the aluminum room.

GENERAL CONCLUSIONS

1. The primary effect of adding aluminum foil insulation to the walls and ceiling of the brick test room was to reduce the inside film coefficients. This was due to the reflective capacity of the foil and resulted in a reduction in radiant heat transfer at the surfaces.

The temperatures of the interior surfaces of the exposed walls were 4 to 5 deg lower, and the temperature of the lower ceiling surface was 5 to 6 deg higher with the reflective paper. The mean air temperature was 0.6 deg lower and the mean globe temperature was 1.2 deg higher in the insulated room than in the uninsulated room.

2. The response to heating or cooling was much more rapid in the insulated room.

3. The addition of the aluminum insulation lowered the coefficient of heat transfer from inside air to outside air from 0.29 Btu per (hr) (sq ft) (deg F) to 0.23. This reduction of 20.7 per cent was reflected in an actual reduction of 20.8 per cent in the heat necessary to maintain comparable conditions.

4. Furthermore, the percentage saving of heat input permitted by an 8 hour period of night cooling was increased by the aluminum insulation. Where 100 units of heat input were necessary with the uninsulated wall, kept continuously at 70 F for 24 hours, only 66 to 67 units were needed to keep the insulated room at 70 F for 16 hours, allowing it to fall to a minimum of 50 F during an 8 hour phase of night cooling.

5. Attention is called to the fact that, when working with surfaces of high heat reflectivity, the thermopile yields results which may be many degrees higher than the actual surface temperature which governs heat transmission outward through the wall. On the other hand, thermocouples on the surface of such a wall may yield values slightly higher than the true temperatures, unless these thermocouples are shielded from direct radiation received from other surfaces.

DISCUSSION

W. H. CARRIER, Syracuse, N. Y.: Has the Effective Temperature been evaluated with and without the aluminum coating on the walls? In view of the radiant heat transferred between walls and thermometers or thermocouples it is important to know how the temperatures are obtained and what the effective temperature is.

G. L. TUVE, Cleveland, Ohio: I do not think we have paid enough attention to the emissivity of brick used on both inside and outside surfaces.

L. E. SEELEY, Durham, N. H.: As there were three or four walls the saving reported by lowering of the night temperatures might be higher than would normally be obtained in a residence.

J. W. MILLER, Lansing, Mich.: Can the information in this paper be used in computing the heat loss of a construction employing thin inside and outside aluminum surfaces?

AUTHORS' CLOSURE: Temperatures reported were operative temperatures which showed the net effect of radiation and convection. Referring to Dean Seeley's com-

ments; the savings reported apply to a two-room one-story building and would be much less in a large building which would also have a much greater storage capacity.

Surface coefficients were found to check values in THE GUIDE of the Society and the results of tests would indicate no change in the present method of computing overall air to air heat transfer coefficients.

In Memoriam 1946

Name	Joined
Walter J. Armstrong, Montreal, Canada	1938
Roy F. J. Becker, Minneapolis, Minn.	1943
Edwin A. Bennett, New York, N. Y.	1929
Tom Brown, Detroit, Mich.	1930
Alfred L. Browne, South Orange, N. J.	1923
Fred C. Burns, Kansas City, Mo.	1942
Carl D. Bushnell, Pittsburgh, Pa.	1921
Ira R. Cummings, Cranford, N. J.	1943
Edwin A. Davies,* Des Moines, Ia.	1944
Otis E. Davis, Scottsbluff, Neb.	1925
William C. Deterling, New York, N. Y.	1937
Thomas Jefferson Duffield, New York, N. Y.	1927
Clayton A. Dunham (*Life Member*) Chicago, Ill.	1911
Harrold English, Los Angeles, Calif.	1930
Ernest J. Febrey (*Life Member*) Washington, D. C.	1903
John B. Feehan (*Life Member*) Lynn, Mass.	1923
Mrs. Olive E. Frank, Paterson, N. J.	1919
William C. Green (*Life Member*) Cincinnati, Ohio	1906
William Nelson Haden (*Life Member*) Trowbridge, England	1902
Charles F. Hauss (*Charter* and *Life Member*) California, Ky.	1894
Frank Henry Hecht, Pittsburgh, Pa.	1930
Martin H. Hirsch, Forest Hills, N. Y.	1938
Norman A. Hollister, Washington, D. C.	1933
Frederick DuBois Ingalls (*Life Member*) Reading, Mass.	1906
Henry G. d'Issertelle (*Life Member*) Tarrytown, N. Y.	1912
Walter Klie (*Life Member*) Cleveland, Ohio	1915
Oliver L. Maddux, Chatham, N. J.	1933
Arthur McGonagle, Pittsburgh, Pa.	1932
Charles W. Miller, Milwaukee, Wis.	1908
Barney Olson,† Chicago, Ill.	1929
William B. Paterson, Ottawa, Ont., Canada	1920
Clyde R. Place, New York, N. Y.	1924
Clarence J. Rice, Milwaukee, Wis.	1923
Otto C. Ross, Indianapolis, Ind.	1944
Wilbur F. Smith, Philadelphia, Pa.	1920
L. A. Stephenson, Kansas City, Mo.	1917
Hugo Theorell, (*Life Member*) Stockholm, Sweden	1902
Carl W. Vollmann, Montreal, Que., Canada	1938
Theodore Weinshank (*Life Member*) Chicago, Ill.	1906
Kenneth C. Woodward, Meriden, Conn.	1944

* Died in 1945
† Died in 1943

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